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ASHRAE JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

How heat flows through glass protected by roller shades page 49

What frost action did to a cold storage plant page 54

Load calculations using pretabulated functions page 67

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If building walls are mostly glass page 75

formerly REFRIGERATING ENGINEERING INCLUDING AIR CONDITIONING AND THE ASHAE JOURNAL

APRIL 1959

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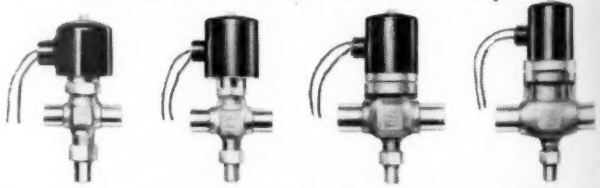

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VOL. 1 NO. 4

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ASHRAE JOURNAL

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SO WHO WANSA LEARN . . . ?

There was a quotation on the wall of the laboratory of Master Empiricist Thomas A. Edison, as we recall it, which read "A man will resort to almost any expedient to avoid the one real labor of thinking".

There, we hold, lies the essence of much of importance that there is to be said today on the subject of education. Our real problem is not the "As", that is everybody; but of the "Gs", those who constitute the hard core of college graduate potential. It is a matter of finding and encouraging those who can and will; of eliminating those upon whom efforts must only be wasted.

Out of an all-embracive "A" number of youngsters there will drop out those who for one reason or another, perhaps several, will never make the grade. Difficulties may include lack of receptivity, aptitudes, inclinations and brain power. Personality traits and inability to utilize the education actually won may deplete the "A" group to produce the "G" group. All this is well known. The trouble arises in failing to discriminate before matriculation and then, almost certainly, pushing laggards on the theory that, out of a certain total, so and so many must graduate.

Facing an unpopular fact, such as that there are foreseeable limits as to who can be taught what, is not pleasing to some educators. It may reduce the prospects for grandiose handouts for elaborate educational plants and structures. It may interfere with personal aspirations and ambitions. It hits squarely at the untenable theory that "all men are equal". It endangers the practice of athletic scholarships. It is politically dangerous. So, instead of realistic appraisals of matriculating talent, we hear more and more feverish pleas for funds, schools, teachers and college graduates in greater and greater quantities.

It can be contended that not half of those who go to college should ever have been admitted; that their being there at all is more of a social accomplishment or a strictly business adjunct, than anything else.

There are those who maintain that what is needed in our country is not a broader and more inclusive educational structure but a far higher and more selective one.

For our part, we are in favor of more forthright thought and discussion of this subject starting with qualifications of the applicant and proceeding directly to the economics of educational accomplishment. It seems sometimes that too many feel that a mere outpouring of community or corporate funds will assure whatever favorable outcome is anticipated.

Further, we think that the time is ripe for just such soul-searching analysis and re-evaluation by us all.

Edward R Searles
Editor

Engineering the temperature...

IN SINGAPORE

Twelve Recold Multizone units air condition the American International Assurance Building. These Recold Multizone units, one on each floor, supply 108,000 CFM of conditioned air for this modern, 34 zone, office building. One standard Recold air conditioning unit cools the executive penthouse.

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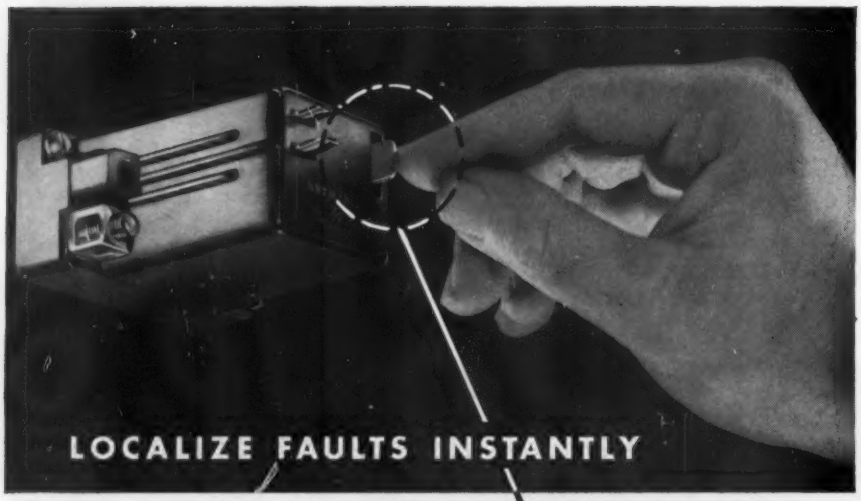

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Widely used (and highly praised) in its standard, non-indicating form, Heinemann's Type C Relay is now available in two new models designed to simplify equipment troubleshooting. The new relays offer the same close-tolerance overload protection, the same temperature-stable tripping characteristics and the same selection of time-delay responses as the standard Type C.

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All models, of course, operate on Heinemann's well-proven principle of hydraulically controlled time delay with magnetic actuation. Tripping is instantaneous on short circuits . . . or on any overload exceeding the fixed instantaneous-trip current value. Effective single phasing protection is provided for polyphase motors. And because the Type C Relay uses no thermal elements, it may be accurately and closely rated to the safe loading limits of the protected equipment — without compensation for ambient temperature effects. "Must-hold" and "must-trip" points are, therefore, reliable values which assure precise, permanent overload protection.

Complete information and detailed specifications are given in Bulletin 5103. Write for a copy.

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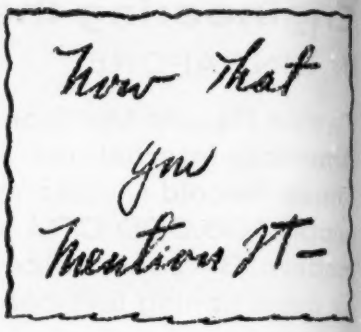

. . . LOOKING FOR A PRACTICAL SHORTCUT

To the Editor:

In reference to the comments by Messrs. Kusuda and Ramsey (March 1959, ASHRAE Journal) on "Predicting Leaving Dry Bulb Temperatures for Air Conditioning Coils" (January 1959, Refrigerating Engineering), I acknowledge that we were wrong, as Mr. Kusuda points out, in attempting to correlate ADP and the *leaving* surface temperature. It appears possible that the ADP is equivalent to some average surface temperature within the coil, but such an average is difficult to determine. Measuring surface temperature with thermocouples, as mentioned by Mr. Kusuda, is risky, because the very presence of the couple at the surface tends to introduce an error in the reading.

Mr. Kusuda's observation that it is "fairly obvious", without involving the kinetic theory of gases, that the cooling curve is independent of face velocity and coil geometry warrants further study. I am not sure of the differential equation to which he refers, but believe that to apply it to a specific coil, it is necessary to introduce dimensional factors.

This observation is also quite interesting in the light of the statement by his colleague, Mr. Ramsey, ". . . convinced . . . these factors (geometry, etc.) do affect the curve". Mr. Ramsey also points out that the effects of the variation of geometry and velocity may have been too small to be apparent. This may well be true, but as he points out, for the variations in actual use, the deviation of the curve is small. Since we were looking for a practical shortcut to the determination of the leaving conditions, I feel a little engineering license might be permissible.

Such license is also assumed in regard to Mr. Ramsey's comment on "fogging". Since we were only concerned with normal air conditioning entering conditions, and not the high humidity, long cooling range situation, described in reference (4) of our bibliography, it seems very reasonable to use the temperature of the leaving surface as the point at which the cooling curve will eventually reach the saturation curve. Any error introduced into the shape of the curve by this assumption will certainly be negligible under normal air conditioning conditions.

ALAN S. DECKER
Vice-President—Engineering
Dunham-Bush, Inc.

Annual ASHRAE Meeting
Lake Placid, N. Y.
June 22-24, 1959

Late news highlights

Freezing poultry Freshness of poultry when it is put into the freezer; wrapping; temperature at which it is frozen and stored; and length of storage are factors which determine the quality of home frozen poultry, report University of Minnesota researchers. In order to prevent freezer burn, wrapping with water-vapor-proof polyethylene bags or heavy duty aluminum foil is recommended. Further, temperature in the freezer should be 0 F or lower and poultry should not be frozen for more than nine months.

Design engineering Fourth annual Design Engineering Conference, sponsored by the machine design division of the American Society of Mechanical Engineers, will feature sessions on choice of materials in design, mechanical aspects of design, and power and control in design. Conference will be held concurrently with the Design Engineering Show in Philadelphia May 25-28. More than 400 companies are expected to exhibit at the show, which is keyed to research and development.

Job Directory Now available, the fifth (1959) edition of Engineers' Job Directory contains information on 1,000 companies, including their plans for expansion, personnel requirements, new products, employee relations and financial standings. Copies of the 72-page directory may be obtained from Decision/Inc., 4601 Red Bank Road, Cincinnati 27, Ohio, for $5.

Award of Merit Cited for "contributions in fundamental and applied research which have greatly aided the development of the frozen food industry," the Western Regional Laboratory of the U. S. Department of Agriculture in Albany received an Award of Merit from the National Association of Frozen Food Packers at its annual meeting in Chicago. Specifically, the award was presented for a comprehensive research project on the extent to which top quality in frozen foods is retained under various temperatures and during different intervals.

Nuclear Congress Dr. Herbert F. York, newly-appointed Director of Research and Engineering for the Defense Department, will deliver the major address at the Nuclear Congress banquet in Cleveland on April 8. The Congress, sponsored by 27 engineering and scientific groups in the U. S. and Canada and by independent atomic technicians to provide the latest information on the use of atomic energy for peacetime purposes, will take place April 5-10.

New filter Philco Corporation is using a new urethane foam for filters of its 1959 line of window air conditioners. This foam differs from other types of urethane foams because of its fully open-pore structure. Unlike other open cell urethane foams which are composed of strands connected by thin membrane-like windows, this material is cited as offering a three-dimensional structure of skeletal strands alone, which reportedly results in a higher porosity and 100% permeability.

Handbook **Handbook of Air Conditioning, Heating and Ventilating,** Clifford Strock, Ed., provides authoritative and useful engineering data in these fields. The book is divided into eight main sections: air conditioning; air handling and ventilation; fuels and combustion; heating and heat transmission; piping and plumbing; motors and starters; mathematical data; and terminology. Emphasizing working data, numerous tables, charts, formulas and graphs useful in solving problems of design, installation and operation are included. Consisting of 1094 pages of data, 598 charts, maps and illustrations and 518 tables, the book is available from the Industrial Press, 93 Worth Street, New York 13, New York. Price is $15.

Longer hours

With orders for packaged products, including air conditioners, considerably higher than last year, the Decatur (Ill.) Works of the York Div of Borg-Warner Corporation will work a two-shift, six day week, according to Henry M. Haase, York president.

Engineering science at Florida State

In order to provide a single curriculum for students wishing to follow careers in research, development and teaching, Florida State University at Tallahassee has organized a Department of Engineering Science, with Dr. Grover L. Rogers as director. Curriculum will be oriented toward mathematics, modern science and theoretical and experimental aspects of engineering.

Twice as large

Newest Alford Refrigerated Warehouse in Corpus Christi will have a capacity of 1,100,000 cu ft, approximately twice the size of the average refrigerated warehouse, and will be 240 ft wide and 350 ft long, larger than a football field. Work has already begun on the warehouse which will serve the South and Coastal Texas.

BRI Meeting

Research directors of six large manufacturers of building products will discuss the research which is being planned for the 1960's and what this will mean in terms of design and construction of buildings in the 1970's at the Building Research Institute's 8th Annual Meeting in Pittsburgh on April 7 and 8. Other technical sessions scheduled for the conference are: sealant for curtain walls; mechanical fasteners in building; a workshop on windows; and a general "look to the future" in the fields of building finance and city planning, home building and building design.

Water conservation

Directed toward industrial people at the plant engineering and upper management level, a short, intensive course on "Industrial Water Conservation," which will explore methods used by industry to cut down on use of water, has been scheduled for June 8 and 9 at the School of Public Health of the University of Michigan.

Curriculum extended

A mathematics liberal-science-oriented curriculum, which was introduced last September in the aeronautical engineering department of the College of Engineering of the University of Detroit is being extended to three additional departments: civil, electrical and mechanical engineering. The curriculum stresses basic sciences, the development of intellectual power, integration of all subjects and mathematics.

NARW Directory

National Association of Refrigerated Warehouses has released its 1959 Directory containing complete information on facilities of its members. Free copies are available to qualified firms from NARW, 1210 Tower Building, Washington, D. C.

Charge it

Mr. Service Club, Inc., a Chicago credit card service organization which provides homeowners with repair and maintenance services on a 24-hour basis announces that it will expand its program by issuing franchises to major market areas in the U. S. and Canada. Presently the club guarantees service in 400 categories of home maintenance and repair and has more than 800 contractors ready to service calls.

Application of fundamentals

In Mechanical Engineering, Second Edition, authors Norman R. Sparks and Charles C. DilLio present theoretic thermodynamic background as well as practical applications in the field of refrigeration. Since the study of refrigeration cycles and systems provides good examples of applied thermodynamics, it is useful to explain principles applicable in other heat power fields. Thus, the text is of potential value not only to those interested primarily in refrigeration, but also for those seeking a book which demonstrates the application of fundamentals. (New York: McGraw-Hill Book Company, 283 pp., $8)

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APRIL 1959

Heat flow

through glass, with roller shades

NECATI OZISIK

In this study, summer heat gains and winter heat losses through a glass-roller-shade combination were investigated for shades of different materials and colors as well as for different type glasses and various glass-shade openings. Included are data related to heat gains for typical combinations in East, South and West orientations and for various hours of the day.

L. F. SCHUTRUM
Member ASHRAE

This investigation of heat flow through glass and roller-shade combinations under summer and winter conditions was under the guidance of the former ASHAE Technical Advisory Committee on Heat Transfer Through Fenestration. Thus, the types and colors of shade materials to be tested were selected at a meeting attended by representatives of the shade material manufacturers and members of the ASHAE Research Laboratory staff. The color and material of the shade and its positioning with respect to the glass were recognized as significant factors influencing the heat flow. Roller-shade materials used for testing included:

1. both sides white, vinyl plastics coated fabric, opaque
2. both sides white, vinyl plastics coated fabric, translucent
3. both sides dark green, vinyl plastics coated fabric, opaque
4. one side aluminum pigmented coated, other side white vinyl plastics coated fabric, opaque
5. one side dark green vinyl plastics coated fabric, other side aluminum foil (i.e. aluminum foil was pasted on one side of the dark green material), opaque
6. both sides aluminum foil, metallic, opaque

Opaque materials did not permit any solar radiation to pass through them, whereas, some solar radiation passed through the translucent material.

Special tests were made at the Laboratory to determine the properties of these materials. The solar reflectance was determined by comparing reflectance of the shade with that of a magnesium carbonate block, using a pyrheliometer. Transmittance for solar radiation of the translucent material was also obtained with a pyrheliometer.

The emissivity of the materials for long-wave radiation was measured using an apparatus as described in Reference 1.

The glasses used included ¼-in. regular plate glass and ¼-in. heat absorbing glass. The properties of the glasses and shade materials tested are given in Table I. Properties of similar shade materials from different manufacturers were within 10 percent of the values given in this table.

The shade was positioned to the calorimeter window with usual brackets in the conventional manner. Spacing between the glass and shade was 5¾ in. with outside type bracket and ¾ or 3 in. with inside brackets (see Fig. 1).

TEST APPARATUS

Except for minor changes the solar calorimeter used in these studies was essentially as described in References 2 and 3. The heat gain through the 42-in. square test window was measured with the inside of the calorimeter maintained at a nominal temperature of 75 F.

Measurements included direct and diffuse components of the incident solar radiation, shade and glass temperatures, outdoor air temperature, air temperature in the space between the shade and glass, and the usual measurements for heat flow determination.

The solar calorimeter is shown in Fig. 2.

EXPERIMENTAL RESULTS

Consider a single glass and a roller-shade combination with solar radiation falling upon the glass

Necati Ozisik is a Research Engineer, L. F. Schutrum is Research Supervisor, both with the ASHRAE Research Laboratory. This paper is the result of research conducted at the laboratory and has been prepared for presentation at an early National Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers.

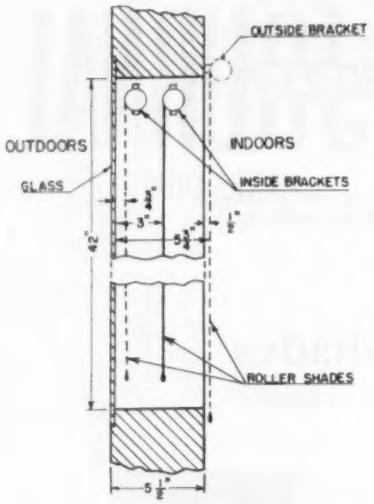

Fig. 1 Positions of the roller-shades

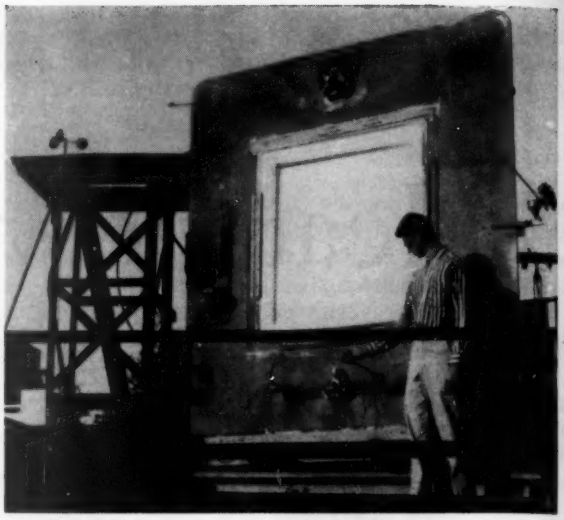

Fig. 2 View of the solar calorimeter

surface. Part of this radiation is reflected, part is absorbed by the glass and shade, and, if the shade material is translucent, part is transmitted through the glass and shade into the room. The amount of energy transmitted can be calculated with reasonable accuracy knowing the transmittances of the glass and shade and the intensity of the solar radiation. The convection-radiation component of the heat gain into the room resulting from the solar energy absorption by the glass and shade, however, is less susceptible of mathematical prediction. The solar energy absorbed by the glass and shade combination increases their temperature until a balance is reached between the rate of heat absorption by the combination and the rate of heat dissipation from the combination by convection-radiation both into the room and to the outdoors.

With energy balance equations for the shade and glass, as given in Appendix A, it can be shown that, for a given glass-shade combination and outside conductance, the convection-radiation component of the heat gain resulting from the solar heat absorption, i.e., CR–U (t_o–t_i), is a function of the solar energy absorbed by the glass and shade combination, i.e. α Igs; and, with no solar radiation both terms are zero, [i.e. α Igs = O, CR = U (t_o–t_i)]. This indicates that, for a given glass-shade combination and outside conductance, if CR–U (t_o–t_i) is plotted against α Igs, the curve passes through the origin. During the tests, however, the outside conductance varied. In order to correlate the test data with the foregoing plot, the data were adjusted to the same outside conductance as explained in Appendix B.

For a 75 F indoor temperature, the CR–U(t_o–75) values from the test data, after being adjusted for an outside conductance of 3* Btu per (hr) (sq ft) (F) were plotted against α Igs, in Fig. 3. As seen from this figure, the data for a given glass-shade combination correlated with a straight line passing through the origin, and the slope of the line represented the fraction

* 3 Btu per (hr) (sq ft) (F) was chosen because experimental data were closer to 3 than the commonly used value of 4.

TABLE I

PROPERTIES OF SHADE MATERIAL AND GLASS

Material	Solar Transmittance	Solar Reflectance	Solar Absorptance	Emissivity for long-wave Rad.
Roller Shade Material:				
Dark Green	0	0.12	0.88[a]	0.90
White Opaque	0	0.80	0.20	0.87
White Translucent	0.22	0.70	0.08	0.89
Aluminum Pigmented	0	0.67[b]	0.33	0.55
Aluminum Foil	0	0.70	0.30[c]	0.07
Type of Glass:				
Ordinary Window[d]	0.87	0.08	0.05	
Regular Plate	0.77	0.07	0.16	
Heat Absorbing	0.41	0.06	0.53	

[a] For design data 0.85 was used.
[b] At 70 degree incidence angle reflectance—was 0.40.
[c] Reference-6 gives 0.19 for a bright aluminum surface. Compared with bright aluminum, experiments indicated an absorptance of about 0.30 for the dull surface aluminum foil.
[d] No test made.
Note: All values are for normal incidence.

TABLE II

K & U VALUES TO BE USED IN EQUATION 1 FOR HEAT FLOW CALCULATION

Shade Material			U BTU	K Values				
Glass Side	Room Side	Type of Glass	(hr) (sq ft) (F)	Incidence Angle of Direct Radiation[a]				Diffuse Radiation
				0-50	60	70	80	
D. Green	D. Green	Regular Plate	0.76	0.52	0.49	0.43	0.30	0.49
D. Green	Al. Foil	"	0.70	0.48	0.46	0.40	0.28	0.45
Al. Foil[b]	Al. Foil	"	0.57	0.22	0.22	0.20	0.15	0.21
White (opaque)	White (opaque)	"	0.76	0.20	0.19	0.17	0.13	0.18
White (Transl.)	White (Transl.)	"	0.76	0.28	0.26	0.23	0.16	0.25
Al. Pigm.	White	"	0.65	0.30	0.34	0.33	0.24	0.30
D. Green	D. Green	Heat Abs.	0.76	0.39	0.38	0.33	0.28	0.38
White (opaque)	White (opaque)	"	0.76	0.24	0.23	0.21	0.18	0.23
White (Transl.)	White (Transl.)	"	0.76	0.29	0.28	0.24	0.20	0.28

[a] Angle between the normal to the surface and the sun's direct beam.
[b] With a bright foil, solar absorptance of about 0.2, K values are about 0.15.
Note: (1) K and U values are adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F) and an inside conductance as suggested in Reference-4.
(2) Glass-shade spacing—3 in.

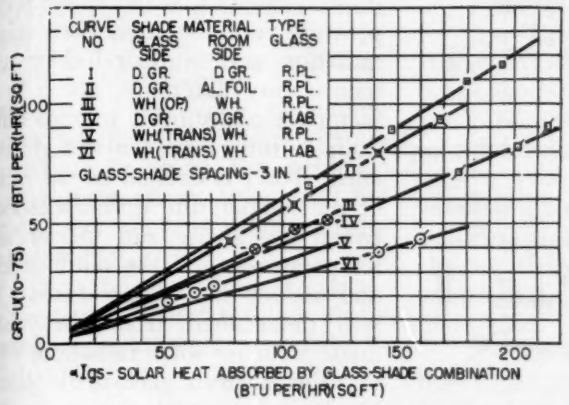

Fig. 3 Convection and radiation heat gain for glass and roller-shade combinations due to solar radiation

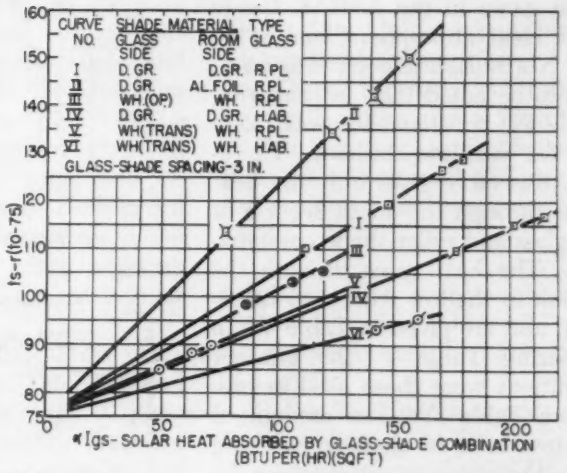

Fig. 4 Roller-shade temperature due to solar radiation

of the solar energy absorbed by the glass-shade combination entering the room as convection-radiation heat gain. Knowing α Igs, U and the slope of the line, the convection-radiation component of the heat gain can be calculated. If the shade material is translucent there is a transmitted component, which can be calculated from the known values of the transmittances of the glass and shade and the intensity of the solar radiation.

For practical purposes, however, it was not so easy to calculate the heat gains in a manner just described. In order to simplify the calculations, the convection-radiation and transmitted components were combined into a single multiplier as explained in Appendix C, and the total heat gain per unit area of the shade-glass combination was reduced to Equation:

$$Q = K.\ I_v + U\ (t_o - 75) \ldots\ldots (1)$$

where K = a factor depending on whether the solar radiation is diffuse or direct, the incidence angle of the direct beam with the glass surface, and the type of glass and roller-shade combination, I_v = the intensity of the solar radiation falling upon the glass surface, U = the overall coefficient of heat transfer.

The K and U values to be used in Equation 1 are tabulated in Table II for the glass-shade combinations tested. K values were calculated as described in Appendix C using the values from Table I and Fig. 3, and adjusted for an inside convection heat transfer as suggested in Reference 4. The U values in Table II are based on the winter tests with an average outdoor air temperature of about 25 F and without solar radiation. However, the U values with higher outdoor temperatures in summer are not appreciably different from the U values given in this table.

The data in Table II are for a fully drawn roller shade, with 3-in. glass-to-shade spacing, and with edge openings of about ½-in. all around the shade.

Therefore, by taking the appropriate K and U values from Table II, heat flow through a glass-roller-shade combination can be calculated from Equation 1, for summer and winter, with and without solar radiation.

The variation of glass-shade spacing from ¾ to 3 in. did not change the heat gain with solar radiation; with 5¾-in. spacing, however, the heat gains were about 5 to 10% higher. With no solar radiation, heat flow was practically unaffected by the variation of glass to shade spacing from ¾ to 5¾-in.

A few tests were made with the edges of the shade sealed to the window frame, with the shade half drawn, and with a fully open shade, and the over-all coefficients of heat transfer for these conditions without solar radiation are included in Table III.

Using Table II values and Equation 1, the summer heat gains through different glass-shade combinations for a 75 F indoor temperature were calculated for East, South and West orientations and for various hours of the day, and the data are presented in Table IV. For these calculations the outdoor

TABLE III

U VALUES FOR A ROLLER-SHADE AND A SINGLE GLASS COMBINATION
(WINTER DATA WITH NO SOLAR RADIATION)

Shade Material		Opening	U[a] BTU (hr) (sq ft) (F)
Glass side	Room side		
		Fully Drawn Shade:	
Cloth	Cloth	½ Inch opening all around the edges	0.76
Al. pigm.	Cloth	½ Inch opening all around the edges	0.65
Cloth	Al. foil	½ Inch opening all around the edges	0.70
Al. foil	Al. foil	½ Inch opening all around the edges	0.57
Cloth	Cloth	All edges sealed	0.53
Al. foil	Al. foil	All edges sealed	0.22
		Half Drawn Shade:	
Cloth	Cloth	½ Inch opening all around the edges	0.88
		No shade	0.95[b]

[a] U values adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F) and an inside conductance as suggested in Reference-4.
[b] Guide value 1.13 adjusted for outside conductance of 3 Btu per (hr) (sq ft) (F).
Note: Cloth shades vinyl coated.

air temperatures and the solar radiation intensities were taken as those given in the ASHAE Guide for a clear atmosphere, and at 40 deg North Latitude on August 1. It is to be noted that, in preparing Table IV, no allowance was made for the window set-back. Therefore, this table gives only the heat gains through the sunlit section of the glass and roller-shade combination. The heat gains through the section in shadow were also calculated and are given in Table V. In preparing Table V, the outdoor conditions were those used in preparing Table IV. The amount of window area in shadow resulting from the window setback can be calculated as suggested in the 1957 Guide.

The shade temperatures for a 75 F indoor temperature, after being adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F) in a manner similar to that used in Appendix B, were plotted against α Igs. In this figure, the ordinate is the shade temperature in the presence of solar radiation adjusted to a zero temperature difference between indoor and outdoor air. This is expressed by $t_s - r(t_o - 75)$, where t_s is the shade temperature resulting from both solar radiation and outdoor-indoor air temperature difference, and r is a factor for adjusting t_s to zero outdoor-indoor temperature difference. The r is defined as the ratio of the temperature differences between the shade and indoor air temperatures, and the outdoor air and indoor air temperatures. It was determined from the tests made with no solar radiation.

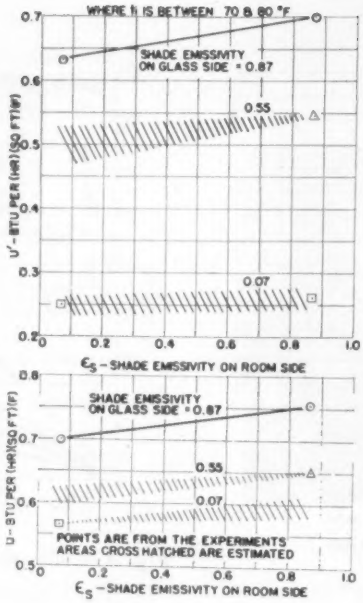
WHERE t_i IS BETWEEN 70 & 80 °F
SHADE EMISSIVITY ON GLASS SIDE = 0.87
0.55
0.07
U' - BTU PER (HR)(SQ FT)(F)
ϵ_S - SHADE EMISSIVITY ON ROOM SIDE
SHADE EMISSIVITY ON GLASS SIDE = 0.87
0.55
0.07
POINTS ARE FROM THE EXPERIMENTS
AREAS CROSS HATCHED ARE ESTIMATED
U - BTU PER (HR)(SQ FT)(F)
ϵ_S - SHADE EMISSIVITY ON ROOM SIDE

Fig. B-1 U and U′ values to be used in Equation B-7, that is, CR = U′

$$\left[\frac{\alpha Ig}{3} + \left(\frac{1}{3} + \frac{1}{h_s}\right)\alpha Is\right] + U(t_o - t_i)$$

where t_i is between 70 and 80 F

For a given glass-shade combination r was constant over a wide range of outdoor air temperatures. With no solar radiation, both $t_s - r(t_o - 75)$ and α Igs are zero, and the curves in Fig. 4 pass through the origin. As seen in Fig. 4, the data for each glass-shade combination correlated with a straight line passing through the origin; and knowing the slope of the line, r and α Igs, the shade temperature can be calculated. For practical purposes, however, α Igs is not an easy term to calculate. Therefore, by means of a procedure similar to that ex-

TABLE IV

HEAT GAINS THROUGH THE SUNLIT SECTION OF GLASS-ROLLER-SHADE COMBINATION FOR 75 F INSIDE TEMP.

Orientation	Suntime	Outside Temp-°F	Heat Gains—Btu per (hr) (sq ft)								
			Regular Plate Glass With Roller Shades			Heat Absorbing Glass With Roller Shades			Glass Only		
			Dark Green	White Transl	White Opaque	Dark Green	White Transl	White Opaque	Ord Window	Reg Plate	Heat Abs
1	2	3	4	5	6	7	8	9	10	11	12
East	6 am	74	77	40	28	57	43	35	128	118	85
	7	75	120	63	45	90	67	56	199	183	135
	8	77	126	67	49	95	71	60	211	192	142
	9	80	114	62	46	88	66	56	187	172	129
	10	83	85	48	38	67	50	44	137	124	94
	11	87	47	29	25	40	31	28	71	67	54
	12	90	23	17	16	20	18	17	33	32	27
South	8 am	77	13	7	6	10	8	7	20	19	14
	9	80	31	18	14	25	19	16	48	44	34
	10	83	51	30	24	42	32	28	78	73	57
	11	87	67	40	32	54	41	37	104	96	75
	12	90	74	44	36	59	46	41	115	107	84
	1 pm	93	71	44	37	58	46	41	110	103	82
	2	94	59	38	33	50	40	36	90	85	70
	3	95	42	29	25	36	30	28	63	59	50
	4	94	26	20	19	23	21	20	37	36	32
West	12	90	23	17	16	20	18	17	33	32	27
	1 pm	93	52	34	29	45	36	33	77	73	61
	2	94	93	57	46	75	59	53	147	136	107
	3	95	125	74	58	99	77	68	204	189	148
	4	94	139	80	62	108	84	73	228	210	164
	5	93	133	77	58	104	81	69	217	202	157
	6	91	90	53	41	70	55	48	144	136	104

Note: (1) Data for regular plate glass can be used for ordinary window glass.
(2) Glass shade spacing—3 in.

TABLE V

HEAT GAINS THROUGH GLASS-ROLLER-SHADE COMBINATION REMAINING IN SHADOW CAUSED BY WINDOW SETBACK (FOR 75 F INSIDE TEMPERATURE)

Orientation	Suntime	Outside Temp-F	Heat Gains Btu per (hr) (sq ft)		
			Dark Green	White Transl	White Opaque
1	2	3	4	5	6
East	6 am	74	10	5	3
	7	75	17	9	6
	8	77	19	11	8
	9	80	21	13	10
	10	83	21	14	12
	11	87	21	16	14
	12	90	22	17	16
South	8 am	77	11	6	5
	9	80	15	10	8
	10	83	19	13	11
	11	87	23	16	14
	12	90	26	19	17
	1 pm	93	27	21	19
	2	94	27	21	19
	3	95	26	21	19
	4	94	24	19	18
West	12	90	23	17	16
	1 pm	93	26	20	18
	2	94	29	21	20
	3	95	32	24	22
	4	94	32	23	21
	5	93	30	22	20
	6	91	23	18	16

Note: For ordinary window, regular plate, or heat absorbing glass.

plained in Appendix C, the shade temperature for a 75 F indoor temperature can be expressed by Equation:

$$t_s = S\ I_v + r\ (t_o - 75) + 75 \quad ..(2)$$

where, S = a factor depending upon whether the solar radiation is diffuse or direct, the incidence angle of the direct beam with the glass, and the type of glass-shade combination.

For the glass-shade combinations tested, S and r values to be used in Equation 2 are given in Table VI.

APPLICATION OF DATA

The summer heat gains through a glass-roller-shade combination can be calculated from the data in Tables IV and V. The former is for the heat gains through the sunlit section and the latter for the section in shadow caused by the window setback. The amount of window area in the shadow can be determined as suggested in The Guide (pp. 320, 1957). It is to be noted that Tables IV and V are for an inside temperature of 75 F, and for outside temperatures as indicated in the table for different hours of the day. For temperature differentials other than those used in these tables, the data may be corrected by adding or subtracting U Btu per (hr) (sq ft) for each degree increase or decrease in temperature differential. The U values are given in Table II.

Although the Table IV data for the heat gains with roller-shades are for regular plate and heat absorbing glass only, the data for a regular plate glass can be used for ordinary window glass. In such a case, heat gains will be about 5% higher with dark green shades and 8% lower with white shades than the corresponding values in Table IV for the regular plate glass.

For outdoor conditions other than those shown in Tables IV and V, the heat flow through a glass-roller-shade combination can be calculated from Equation 1, and the K and U values for this equation can be taken from Table II.

For a given outdoor condition, the shade temperature can be calculated from Equation 2, and the S and r values for this equation can be taken from Table VI.

Example: A westerly oriented, 3-ft wide x 5-ft high regular plate glass window is covered by a fully drawn dark green roller-shade spaced 3 in. from the glass. The window setback is such that at 3 p.m. 18% of the surface of the glass is in the shadow. Calculate the heat gains at 3 p.m. from Tables IV and V for (1) 75 F indoor and (2) 80 F indoor temperatures. (3) Calculate the shade temperature at 3 p.m. for 75 F indoor temperature.

Given: The Guide values for the solar radiation intensities at 3 p.m. are: Direct (normal) 267 Btu per (hr) (sq ft), diffuse on a West wall, 35 Btu per (hr) (sq ft), incidence angle on a West wall 47.7 degrees, and cosine of the incidence angle 0.673.

Solution: The window area in shadow $3 \times 5 \times 0.18 = 2.7$ sq ft and the sunlit area $3 \times 5 - 2.7 = 12.3$ sq ft.

(1) For 75 F indoor temperature: The heat gain for the sunlit section (from Table IV, column 4) $12.3 \times 125 = 1537.5$ Btu per hr; for the section in shadow (from Table V, column 4) $2.7 \times 32 = 86.4$ Btu per hr. The total heat gain $1537.5 + 86.4 = 1624$ Btu per hr.

(2) For 80 F indoor temperature: Table IV and V data for 3 p.m. are based on the air temperature difference $95 - 75 = 20$ F, whereas for this example it is $95 - 80 = 15$ F. Allowing U = 0.76 (from Table II) Btu per (hr) (sq ft) the correction per difference in temperature differential, the reduction in the heat gain is $15 \times 5 \times 0.76 = 57.0$ Btu per hr. Hence, the heat gain is $1624 - 57.0 = 1567$ Btu per hr.

(3) Shade temperature for 75 F indoor temperature: The shade temperature can be calculated from Equation 2 by taking S and r values from Table VI. For the direct solar radiation

(*Continued on page 103*)

TABLE VI

S & r VALUES TO BE USED IN EQUATION-2 FOR SHADE TEMPERATURE CALCULATIONS

Shade Material		Type of Glass	S Values					
			r	Incidence Angle of Direct Radiation				Diffuse Radiation
Glass Side	Room Side			0-50	60	70	80	
D. Green	D. Green	Regular Plate	.27	.24	.22	.20	.14	.22
D. Green	Al. Foil	"	.31	.38	.36	.31	.22	.35
Al. Foil[a]	Al. Foil	"	.22	.35	.33	.29	.20	.33
White (Opaque)	White (Opaque)	"	.27	.11	.10	.09	.07	.10
White (Transl.)	White (Transl.)	"	.27	.06	.06	.06	.05	.06
Al. Pigm.	White	"	.24	.15	.17	.17	.12	.15
D. Green	D. Green	Heat Abs.	.27	.17	.16	.14	.12	.16
White (Opaque)	White (Opaque)	"	.27	.11	.10	.09	.08	.10
White (Transl.)	White (Transl.)	"	.27	.09	.09	.08	.07	.08

Note: S & r values are adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F) and an inside conductance as suggested in Reference-4.
[a] With a bright foil, solar absorptance of about 0.2, S values are about 0.25.

TABLE VII

REDUCTION IN HEAT GAIN BY FULLY DRAWN ROLLER SHADES

Type of Glass	Heat Reduction by Roller Shades Percentage (a)					
	Regular Plate			Heat Absorbing		
Type of Roller Shade	Dark Green	White Transl.	White Opaque	Dark Green	White Transl.	White Opaque
South (b)	30	56	64	28	43	49
West (c)	33	60	68	32	47	53

[a] Percentage of heat gain through unshaded glass.
[b] From 8 am to 4 pm.
[c] From 12 noon to 6 pm.
Note: Data are for a window with no setback and based on table 4 values.

What frost action did

to a cold storage plant

Here is a case history of the damage done by frost action in the ground under a specific cold storage plant together with the remedial measures suggested. Such studies are infrequent. The damage seldom manifests itself fully in less than five years. That information pertinent to the case is often lost to review in the interval.

It would appear that in many instances designers either do not appreciate fully the dangers involved in building on frost-susceptible soils or fail to provide adequately for the prevention of freezing of such soils in their designs.

J. J. HAMILTON D. C. PEARCE N. B. HUTCHEON

In Cornwall, Ontario, the Stormont Cold Storage Co-operative operates a conventional locker plant having office and processing rooms, sharp freezing facilities, holding rooms and rental lockers in a large locker room. The building is a single story reinforced concrete structure founded on shallow spread footings. It has a steam boiler in a basement boiler room for heating the office and retail store area in the front of the building. Refrigeration is by a direct expansion system with pipe coils suspended from the roof of the cold rooms. The water-cooled compressors are located in a semi-basement machinery room also located in the front of the building. The main floor plan of the original plant is shown in Fig. 1.

Fig. 2 shows a typical section through the floor slab, a column and footing, and also shows a soil profile representative of the subsoil adjacent to the floor slab and footings. The floor slab is of sandwich construction with two 3-in. reinforced concrete layers separated by 6 in. of cork insulation. The tiles, as shown, were introduced as a modification to the original design. They were intended to provide a ventilating duct system, below the slab, to assist in preventing freezing of the ground. Hollow structural clay tiles were laid end to end, in parallel rows terminating in 9 x 9-in. headers formed in the concrete at either end. It was intended that air from the boiler room should flow through this system and be discharged into the compressor room. This system was ineffective since no fan was provided. It did, however, serve a useful purpose as later described.

At the time of construction the site was observed to be generally low lying, having poor natural surface drainage and a higher groundwater table. According to the contractor the original surface of the ground was from 1 to 2 ft below the present outside grade as shown in Fig. 2. Individual excavations were made for interior column footings and perimeter wall footings. Some difficulty with groundwater in the deeper excavations for wall footings was noted during the forming and concreting operations, the water table then being from 3 to 4 ft below the original ground surface. After placing of the concrete for the footings, the excavations were backfilled with "pit-run" gravel to the required grade of the tiles and the bottom of the slab. This gravel is locally used for concrete aggregate and was described as being "clean and of good quality." Owing to the method of placing the footings, the present soil profile below the building may not have a regular horizontal stratification. In areas where no excavation for footings was required it is assumed that the gravel backfill was placed directly on the original ground surface while over the footings the gravel was backfilled from the top of the footing to the bottom of the floor slab.

The plant was put into service in 1951 with temperatures of –20 F being maintained in the freezing room and –5 F in the locker room. Later the wall separating these two rooms was damaged by heaving and was removed. Thereafter both rooms were operated at –10 F.

The operation of the plant proved satisfactory until the spring of 1956 when heaving of the floor

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J. J. Hamilton and D. C. Pearce are Research Officers, N. B. Hutcheon is Assistant Director, Div. of Building Research, National Research Council of Canada, Ottawa, Canada.

This paper links closely with those broader aspects of the problem reviewed by authors Pearce and Hutcheon as "Frost Action Under Cold Storage Plants," REFRIGERATING ENGINEERING, October 1958.

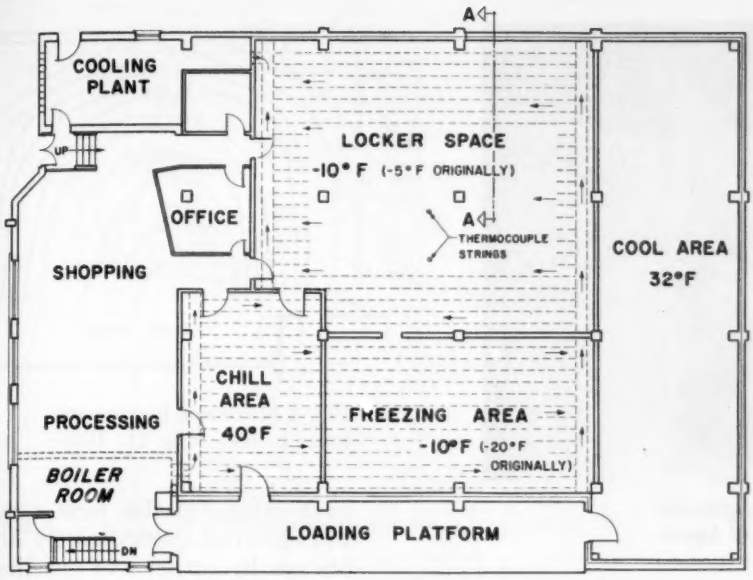


Fig. 1 Ground floor plan

slab was first noticed in the locker area. Subsequent investigation showed the tile subfloor ventilating system to be blocked. This was cleared and a propeller fan was installed in the compressor room in an attempt to induce circulation. The fan was operated from May until December 1956. Frost heaving continued rapidly, however, and by January 1957, when the problem first was brought to the attention of the Division of Building Research, the floor slab had sustained a maximum deflection of 11 in. The column footings were also displaced upwards (though to a lesser degree), resulting in a doming of the reinforced concrete roof slab.

The first phase of the study by the Division of Building Research was **to install thermocouples beneath the floor slab.** Two "strings" of thermocouples were installed through holes drilled in the floor with a rotary air drill. Each string consisted of copper-constantan thermocouples spaced at 1-ft intervals, encased in a 9-ft polyethylene tube which was packed with sand and sealed to prevent moisture penetration. One string was placed near the footing of an interior column and the other at approximately the center of the bay where maximum heaving had occurred.

At the time of installation of the thermocouples it was found that the subsoil was frozen to a depth of about 8 ft 8 in. in hole No. TA and to 8 ft 10 in. in hole No. TB. It was also observed that the ground temperatures measured in both holes corresponded quite closely at any given depth indicating that the columns had little influence on the temperature distribution. Subsequent observations in April and June 1957 indicated that the depth of freezing had advanced beyond the 9-ft depth.

In addition to subsurface temperature measurements, periodic level surveys of the floor and the roof were made. By June 1957, the deflection experienced by the slab had reached the proportions illustrated in Fig. 3. The extent of the heave is strikingly reflected by this row of lockers which were located near the region of maximum heave. The rather alarming crack to be noted at the base of the column is of no structural significance since only the hardboard covering the cork insulation has been damaged. It resulted from a combination of lateral movement and differential vertical movement of the floor and column. This column was the only one in the building to suffer other than vertical movement. The upward movement of the structure caused this column to be moved nearly 5 in. from the vertical.

Fig. 4 shows the floor heave contours for the worst condition. These contours outline the general symmetry of the heaving with the point of maximum heave being near the dimensional center of the freezing space.

Typical soil profiles have been obtained from the two borings made through the floor and from several others around the building perimeter. Examination of the grain-size distribution curves for the soils below footing level shows them to be highly frost susceptible according to the commonly accepted criteria of Casagrande (1). According to these criteria, the susceptibility of soils to frost heaving is partially dependent on the uniformity of grain sizes. Fairly uniform soils, such as the thin sand stratum found slightly below the footings, are considered susceptible if they contain at least 10 per cent of grains

Fig. 2 Section A-A. Typical floor slab, columns and footings

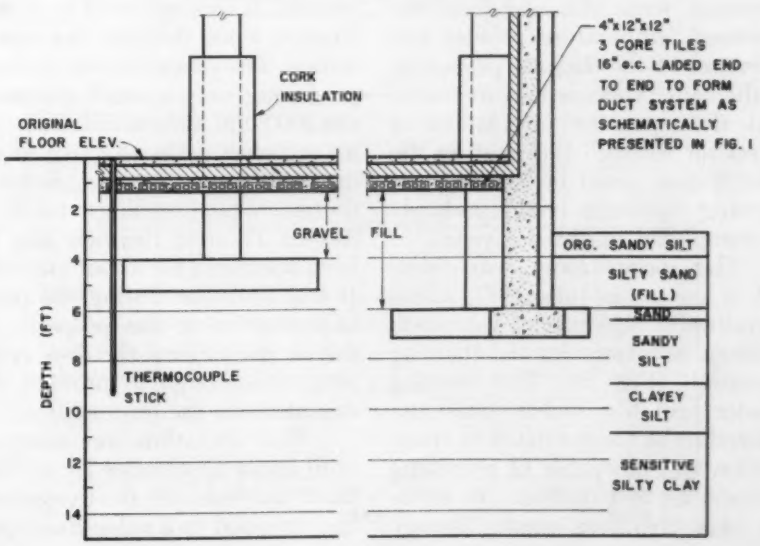


Fig. 3 Interior view of refrigerated locker room showing extent of heave

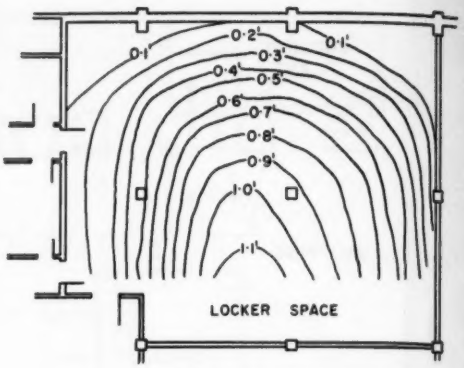


Fig. 4 Maximum heave contours in feet June 11, 1957

smaller than 0.02 mm. This particular sand is on the borderline but might be considered safe. The remaining samples from lower levels showed well-graded (mixed grained) grain-size distributions and their percentage of sizes less than 0.02 mm. were many times the required 3 per cent for frost heaving. Frost heaving does not depend entirely on grain-size distribution since water and freezing conditions must also exist. With the high groundwater table facilitating continuous moisture migration to the freezing front at any depth below the footing level and the comparatively slow continuous heat extraction, a highly favorable condition for frost heaving existed in these soils.

After several discussions with the operators of the plant **it was decided to attempt to thaw the underlying subsoil.** Preliminary estimates of the amount of heat necessary were obtained from the observed temperature profiles and estimates of the thermal properties of the soils. Calculations indicated that if a positive circulation of warm air through the existing tile duct system could be assured, the thawing operation could probably be completed in about a year.

The locker room was taken out of service in June 1957, which permitted completion of the modifications necessary for the thawing operation (Fig. 5). The existing header and tile system was considered to be too restricted in cross-section and incapable of providing uniform air distribution. An opening was therefore made through the floor into the header duct at one end of the room to provide means, in addition to the existing openings to the boiler and compressor rooms, for entry of warm room air. An opening was cut in an outside wall to allow warm summer air to enter the room. The continuous header at the other end of the room was broken into at two points where it was then blocked with mortar to form three separate sections. An opening was then made in the floor over the center of each of these sections and a new duct system and fan installed to draw air from them.

The duct system was designed for 1000 cfm from each of the three sections. A centrifugal fan having a capacity of 3000 cfm at 1 in. static pressure was ordered, but in the meantime the existing propeller fan was used on the new duct system to draw air through under the floor and to discharge it to the outside. It was operated until early August, when the new fan was installed. The propeller fan, although producing only a small portion of the 3000 cfm airflow called for, was nevertheless quite effective in producing thawing as was shown by thermocouple readings taken on August 17 after the new fan had been operating for about one week. It was said that during the period of operation of the propeller fan the air drawn from the floor system was cold enough to produce condensation on the duct system.

This operation was continued until about September 15, at which time the outside air temperature had dropped to a value inadequate for heating. At this time a steam unit heater of approximately 70,000 Btu per hr output was installed in the locker room so as to deliver most of its heat directly into the intake opening in the upstream header. The fan system was modified to exhaust into the storage space. This recirculation was continued until the thawing was complete. The air temperature in the duct system at the center of the floor during this heating operation was 80 F to 85 F. Temperatures of 50 F to 60 F were achieved earlier when circulating outside summer air.

During the thawing operation groundwater table measurements were made with a wellpoint-type gauge installed in one of the soil sampling bore holes located outside the limits of the building but as close as possible to the area of maximum heave. Since the bearing capacity of the soil could be drastically reduced by an increase in moisture content, it was thought desirable to observe any effects on the water table during the thawing of the ice layers in the soil. A dangerous situation could have existed had thawing taken place at a rate faster than drainage could be achieved naturally through the upper sandy and silty layers. It is interesting to note that during the summer and early fall period, the groundwater table dropped slightly, as measured by the wellpoint, reflecting the general seasonal trend for open field conditions.

During the subsoil heating operations thermocouple readings were taken periodically. The first 6 ft of soil thawed rather quickly, the rate of thaw decreasing somewhat below this depth. This is depicted in Fig. 6. As can be seen

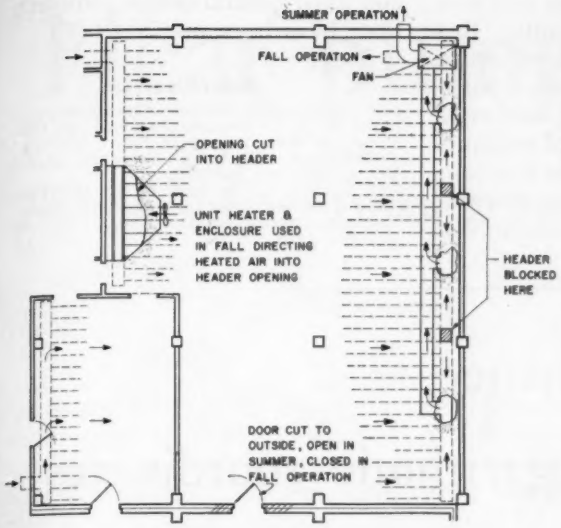

SUMMER OPERATION
FALL OPERATION
FAN
OPENING CUT INTO HEADER
UNIT HEATER & ENCLOSURE USED IN FALL DIRECTING HEATED AIR INTO HEADER OPENING
HEADER BLOCKED HERE
DOOR CUT TO OUTSIDE, OPEN IN SUMMER, CLOSED IN FALL OPERATION

Fig. 5 Modifications for thawing operation

from the maximum heave curve the rate of heave and depth of frozen soil show a roughly linear relationship. This would imply that throughout the frost-susceptible soil, the thickness of ice varied from 3 to 4 in. per ft of frozen soil. From an examination of the heave experienced by the roof slab it appears that the heave below the footings would be less than 2½ in. per ft of frozen soil. This difference may be due to the effect upon the ice formation of increased bearing pressures under the footings.

By late December 1957 the heave had completely subsided (well in advance of the predicted time because of the high air temperatures produced by the steam coil) and the floor slab and interior footings began to experience settlement. By early March 1958 the floor slab and column footings reached equilibrium conditions. Fig. 6 shows vertical displacements of the floor in the region of maximum heave, which was logically the region to suffer the greatest settlement after thawing also, at different times during the observation pericd.

After thawing a maximum differential settlement of 2½ in. was measured. The roof slab was also subjected to a similar deflection due to the settlement of the interior columns following thawing. Since the roof and floor slabs were subjected to equal subsidence, the cause of the final differential settlement was believed to be deep-seated, i.e., at depths below the footing elevation and therefore in "undisturbed" soils below the limits of excavation.

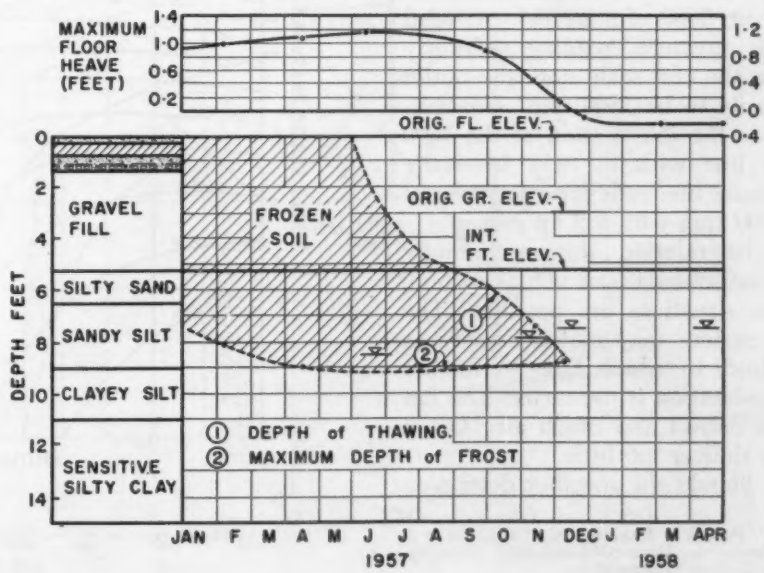
The volume of water required to form 15 in. of ice lenses over the large area affected by frost heaving would be considerable and due to decreasing permeability of the subsoil with depth, the amount of inward seepage towards the freezing front would decrease as the frost line penetrated to greater depths. It is felt that the demand for moisture which was created by the pore water tension induced by ice-lens formation may have been satisfied partially by water from the lower layers of the silty and clayey soil, thus causing a volume reduction in this unfrozen soil. This consolidation, induced by tension in the pore water of the soil, is an irreversible process in clays with a structure which can be altered easily by stress changes.

Inspection of the building following completion of the thawing operation revealed it to be in surprisingly good condition. The interior column that had been displaced from the vertical returned to its original position. Only hairline cracks were seen in the floor slab. The depression in the floor slab created no serious problem from the standpoint of future use of the area and could be eliminated with the application of a wearing course of concrete. The depression in the roof slab, which allowed some ponding of rainfall on top of the asphalt-covered insulated roof can also be easily remedied. Considering the extreme differential movement that had occurred the recovery was quite remarkable.

The owners were pleased with the rehabilitation of the building since they had been able to use the two rooms for dry storage during the process. The whole operation cost less than $1,000. The cost of the floor alone was estimated to be about $10,000 so any project involving removal of the floor for thawing would have been quite costly. At one time serious consideration had been given by the owners to abandoning the storage area.

In the opinion of the authors the building can again be used for

Fig. 6 Depth of frozen soil and corresponding maximum floor heave conditions



low temperature storage provided an adequate subsoil heating system is installed. This study has indicated that warm summer air would be an adequate source of heat and could be circulated by the fan used in the thawing operation. The future use of this area has still to be decided by the owners.

This study affords an excellent illustration of the difficulties that can occur if potential frost action beneath an artificially cooled building is not considered seriously. Fortunately, the remedial measures were highly successful in this case. The Division of Building Research is keenly interested in this problem and some general design considerations have been published (2).

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Fan rpm related to performance by logarithmic coordinates

C. W. POLLOCK

As illustrated in the accompanying figure, it is possible to simplify greatly the analysis of fan performance by plotting constant rpm curves on logarithmic coordinates so that the system characteristic becomes a straight line. Superimposing constant hp curves upon the constant rpm curves permits determining fan selection from a single point simultaneously relating air quantity, fan outlet velocity, total static pressure, hp, rpm and relative fan efficiency — the point of maximum efficiency being the peak of any constant rpm curve.

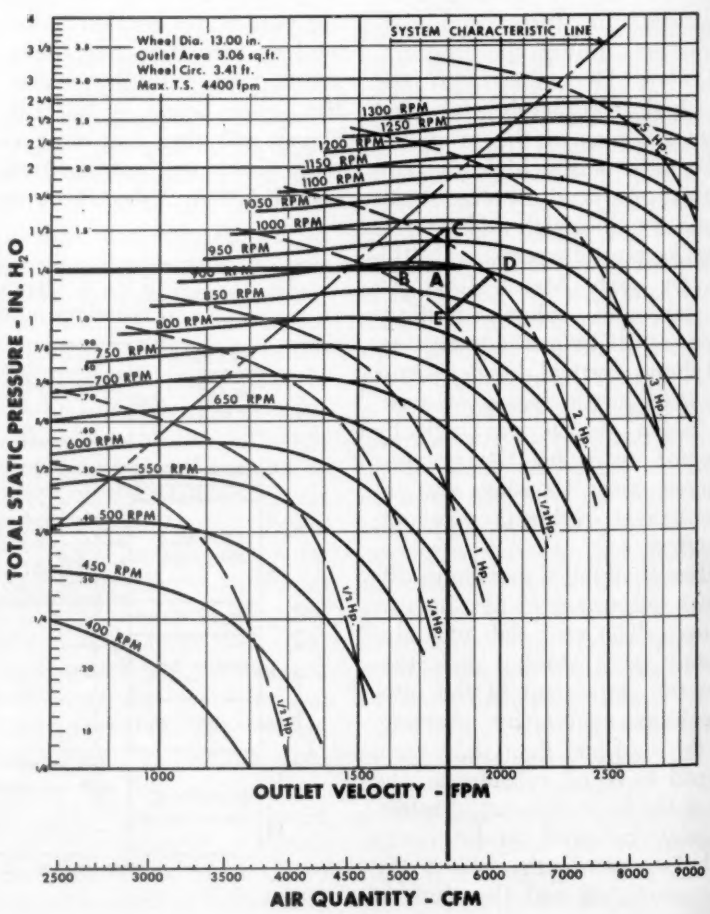
Points of maximum efficiency fall on a line which, for all practical purposes, is parallel to a typical system characteristic line.

For example, where a preliminary selection calls for a fan to deliver 5500 cfm at 1¼ in. total static pressure, reading vertically from the cfm scale note the outlet velocity to be 1800 fpm. Continuing to the intersection of the 5500 cfm line with the 1¼ in. static pressure line calls for a fan to turn at 900 rpm with a 2 hp motor.

In relating this preliminary fan selection to an actual system, three situations are possible. The fan system may match exactly as desired, in which case no further consideration is necessary. The fan may deliver too much air. Or, it may deliver too little.

Should the specifier determine that not enough air is being supplied, a correction in the fan rpm becomes necessary. Assume the fan to deliver only 5000 cfm instead of 5500. Knowing the rpm to be 900, Point B., at 900 rpm and 5000 cfm can be located. Then draw line BC from Point B., parallel to the typical system characteristic, to intersect the 5500 desired performance at Point C. Point C. then indicates the fan should turn at 1000 rpm and probably a 3 hp motor would be required.

In a similar manner, it is also possible that if, in operation, it has been determined that the system is delivering 6000 cfm, instead of 5500 as desired, one could establish Point B. and find Point E., thus determining that the fan speed should be reduced to 800 rpm and probably 1½ hp motor would be sufficient.



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Heat pump performance

for package air source units

W. A. SPOFFORD

Air source heat pumps serve the dual purpose of cooling in summer and heating in winter. The advantages of heating with the heat pump depend to a great extent on its ability to add heating to its cooling function with a minimum of additional equipment and services.

The modifications to add the heating function to electrically operated mechanical refrigeration cooling equipment consist of a switchover valve to reverse the flow of refrigerant, automatic defrost means and other suitable controls for year-round operation.

For economic reasons, the heat pump unit is usually sized to meet the cooling requirements of the installation, and hence bears relatively little direct relationship to the heating needs of the structure. In addition to normal structural heat loss, the heating performance of the system must consider the requirements for heat during defrost and for rapid pick-up if required by the application. Where the total heating requirement exceeds the heating capacity of the unit, some form of supplement is selected—usually electric resistance heat. In residential applications most installations require supplementary heat even though its total heating season usage may be quite small.

Packaged air source heat pumps may be self-contained or of the split system type with indoor and outdoor sections designed to be used together. For simplicity they normally have a single one-stage compressor, and this type is considered here.

The performance of this type of equipment will be discussed in terms of the variables that affect its performance. Since this performance changes with the weather, it is necessary also to consider the average performance for the season, as well as the extent and duration of the extremes.

W. A. Spofford is with the Air Conditioning Dept., General Electric Company. This paper was presented as "Performance of Packaged Air Source Heat Pump Units" at the Symposium on Heat Pump Performance at the 65th Annual Meeting of ASHAE in Philadelphia, Pa., January 26-29, 1959.

TYPICAL PERFORMANCE CHARACTERISTICS

The useful output of the heat pump is the cooling effect within the building in the summer and the

Fig. 1 Relative input and output, heating and cooling

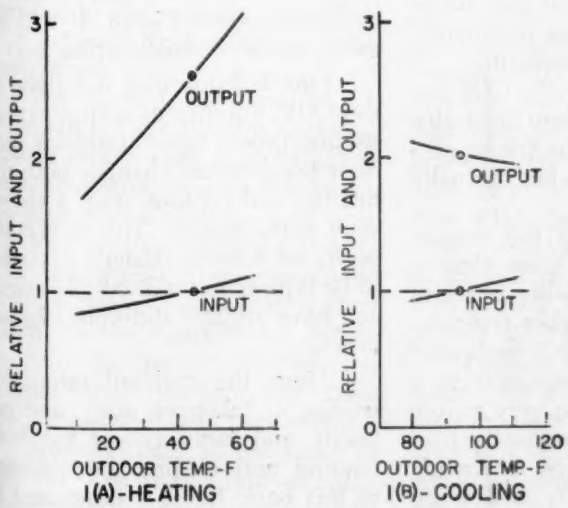
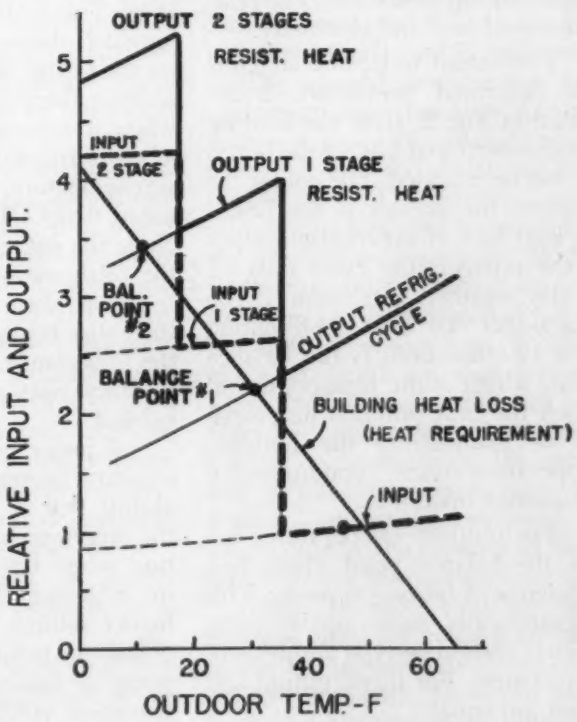


Fig. 2 Heating capacity vs requirements



heating effect in the winter. The input, in either case, is the electrical input required to run the equipment.

The importance of the heat pump arises from the fact that the output always exceeds the input. The output for heating is some 2½ times the input, Fig. 1a; the output for cooling is typically double the input, Fig. 1b. The input and output both change with outdoor temperatures in the manner illustrated. Both diagrams have been drawn in terms of an input of unity at the standard outdoor rating temperature. This is 95 F for cooling and 45 F for heating. This does not mean that the actual inputs are the same at the rating temperatures for heating and cooling in a particular machine. The important matter to notice now is the relationship of output to input for the heating and cooling functions.

Heating output decreases with decreasing outdoor temperature. This is unlike a fossil fuel fired heater in which the output is essentially unaffected by outdoor temperature. As a consequence of this, the over-all economics of heat pump heating are relatively more advantageous in the South or in buildings in other areas where the cooling and heating requirements tend to match the inherent performance characteristics of the heat pump. These effects have their influence on top of the effect of relative cost of fuel and electricity.

The output to input characteristic described previously is repeated in Fig. 2. Here the heating requirement (heat loss) of the building has been added. The colder the weather, the greater is the building heat loss. However, the output of the refrigerating cycle falls off as the weather gets colder. The crossing of the lines representing these two functions is the balance point, which is the temperature at which the heat pump is just capable of maintaining the building temperature under conditions of continuous operation.

For outdoor temperatures below the balance point, there is a deficiency of heating capacity. This is customarily made up by using electric resistance type supplementary heaters. For these, output and input are equal.

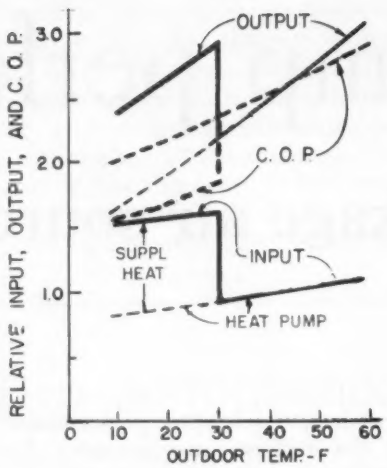


Fig. 3 Coefficient of performance (heating) with supplemental heat

A heat pump of larger capacity relative to the building heat loss could be used, thereby shifting the balance point to the lowest expected temperature. In practice, however, a heat pump with a refrigeration cycle of adequate size for cooling is selected, and for the short periods in which there is a deficiency for heating, it is more economical to use supplementary resistance heaters for packaged size equipment. From this point on, a complete heat pump system consisting of the refrigerating system, including appropriate supplementary heaters, will be considered.

The ratio of the output to the input is the Coefficient of Performance. This COP is plotted in Fig. 3, which shows not only the trend for the refrigeration cycle itself, but also the abrupt drop this COP takes when it represents the composite of the refrigerating equipment and supplementary heat. It should be noted that COP is used here to include the entire system, not merely the refrigeration equipment.

This abrupt drop in COP for the entire heating system illustrates the importance of using the supplementary heat only when actually needed.

A major need for the supplementary heater is to give pick-up ability, that is, the ability to warm the structure to a higher temperature when the desire is indicated by adjusting the thermostat to a higher setting. By the very nature of balance point operation, the heat pump at this condition is already operating continuously and is incapable of raising the building temperature.

Actually, the balance point does not occur at a fixed temperature. It changes with changes in selection of indoor temperature. It is raised by the effect of wind which can greatly increase the heat loss from the building. It can be appreciably lowered during the day when the sun assists in heating the building.

For greatest operating economy then, the refrigeration equipment should operate at all times when heat is required and should operate alone at or above the balance point temperature whatever that may be at the time.

To accomplish this optimum condition, the supplementary heat is best put, first, under the control of an outdoor thermostat that permits such heat only when the outdoor temperature is sufficiently low, and secondly, under control of the second stage of a 2-stage indoor thermostat which will call for this additional heat only when actually needed.

The supplementary heat may consist of two or more units, each successively cut in at lower outdoor temperatures by individual outdoor thermostats suitably adjusted. In this way good operating economy can be obtained.

These considerations show that the COP for the system changes not only with the weather, but with the number of stages of supplementary heat used and with the type of controls.

What is the average COP? The useful value is the ratio of the heat output for the entire season to the heat-equivalent of the electrical input for the season. This is called the Performance Factor. It will be given further consideration later.

One is interested not only in the COP, but also in output capacity and power input trends as outdoor temperature changes both for heating and cooling. Fig. 4 shows such performance. This is a composite of several designs in order to be typical and individual designs can have slightly different characteristics.

Here, the standard rating for cooling is taken as unity and capacity and input power for both heating and cooling are referred to this base. Input is illustrated in

the same units as output and is accordingly the heat-equivalent of the electric power supplied.

Cooling output depends on indoor wet-bulb temperature as well as dry-bulb outdoor temperatures, and a family of curves is shown spanning the typical operating range. Similarly, there is a family of curves representing the spread of indoor temperatures that may be of interest.

Holding an indoor temperature above 70 F reduces the COP. Not only this, but the higher indoor temperature increases the building heat loss. Thus, both factors combine to cause higher operating costs. The indoor temperatures selected by various building occupants can thus have appreciable effect on annual operating costs.

ESTABLISHING COP

Typical heat pump performance has been illustrated. What are the prospects for improvements?

Perhaps the situation can be put in best perspective by considering design variables in a certain order that defers discussion of the refrigerant cycle and the compressor to the last.

Fig. 5 makes this approach in terms of temperature patterns, an output held constant at unity, and the input and COP possibilities stated in a series of steps from the ideal to the actual.

The limitations of the Carnot Cycle are evident here. Carnot first stated this law of nature which, in terms of reference for heat pump heating, is that the maximum conceivable coefficient of performance for heating is the ratio of the absolute temperature at which heat is rejected to the temperature difference between the heat sink and the heat source.

For cooling, the Carnot COP is the ratio of the absolute temperature at which heat is absorbed to the temperature difference between the heat source and the heat sink.

Put another way, the Carnot COP is the maximum conceivable ratio of useful heating or cooling output to the input which has to be paid for, namely the electric power. Input obtained free from the atmosphere is not included.

Now consider in the series of steps of Fig. 5 the typical situation where the indoor temperature is 70 F and the outdoor temperature is 45 F.

Step I. This is the ideal where the Carnot COP is attractively high at 21.2.

Step II. Air must be pumped to distribute the heat indoors, say 100 F. A separate air stream must be moved over the outdoor coil and is cooled to say 35 F. These new temperature limits drop the Carnot COP to about 8.6.

Step III. It is necessary to have heat exchangers and these require an additional temperature spread to transfer the heat. This drops the Carnot COP to about 7.1.

Step IV. Fan power is necessary to move the two air streams. This is about 10 percent of the output. About half of this is for the indoor fan and is useful for heating, but at a COP of 1.00. The other half, for the outdoor fan, is usually not recoverable. Together these cause the second largest drop in COP to about 4.2.

Step V. To provide flow of refrigerant through heat exchangers, pipes and switchover valves, there are pressure drops with corresponding refrigerant saturation temperature drops. These effects are not large, but drop the COP to about 4.

Step VI. The Rankin Cycle (evaporating, compressing isentropically, condensing, re-expanding a fluid) is a cycle suited to use of a volatile refrigerant, but it only approximates Carnot efficiency. Use of an actual refrigerant is only about 80% as efficient as the Carnot Cycle between the same temperature limits. After adjusting again for fan power the COP is now 3.5.

Step. VII. The customarily used hermetically-sealed refrigerant motor compressor puts the motor losses into the refrigerant before compression. Motor losses must also be considered in the input. Assuming an otherwise ideal compressor (isentropic compression) now results in a COP of about 3.0.

Step VIII. Actual overall COP, all things considered, typically drops to about 2.6 for the indoor temperature of 70 F and outdoor temperature of 45 F for this example.

Further comment on design trends for each of these steps is now in order.

Step I. The ideal Carnot COP will never be remotely approached.

Step II. The biggest drop in COP is due to having reasonable air quantities. Indoor air flow is established for cooling at typically 400 cfm per ton of cooling. The outdoor air flow is twice this, or somewhat less. Air quantities affect costs other than in the heat pump itself, such as for ducts, outlets and grilles throughout the building. Air quantities are unlikely to change much, since industry has voted for existing quantities for many years as being economical. The air quantities, external ducts and internal passages through the heat pump determine motor size and power for fans.

Step III. Heat exchangers are a major element of cost. To progressively increase COP requires large increments of heat exchanger for ever smaller gains in COP. Infinitely large exchangers would increase the COP from the 7.1 to 8.6 level shown, if air flows are unchanged. Doubling heat exchanger size would go about three-quarters of the way, that is, from 7.1 to 8.2.

Step IV. Fan power is of major importance dictated by many economic considerations. It accounts for about 20% of operating cost. The fan power does not change with the weather. Sometimes fan power is omitted from consideration of COP and only the compressor power is considered. This is really a misuse of the term COP since input should include all operating elements of the unit such as compressors, outdoor and indoor fans and controls.

Steps II, III, and IV, collectively. The selected air quantities, heat exchanger sizes, fan sizes and fan motor sizes, with related equipment, account for about one-third of the cost of a packaged heat pump when sizes are dominated by the cooling requirements. Within limits, the cost and size of some of the items can be adjusted at the expense of the others with moderate effect on overall cost and performance. This gives the designer latitude for standardizing parts, making full use of standard size motors and the like. However, as explained already, modest improvements in COP come at the expense of large increases in these components collectively. In mild climates these items as sized for cooling are adequate. For increasingly severe heating requirements, improved COP is desired but first costs must then

rise. This is the place for exercise of design and marketing skill to obtain the most practical answer to meet today's needs in the market place.

Step V. Decreases in performance due to piping and switchover valve losses are inevitable, but are moderate and relatively easy to evaluate for cost vs performance.

Step VI. Many factors govern the choice of refrigerant, but there is only about 10% variation in the Rankin efficiency among those likely to be used.

Step VII. The advantage of sealing the compressor motor inside the compressor shell outweighs the disadvantages of using the refrigerant to cool the motor.

Step VIII. Actual performance is the net result of all design factors, with only the major ones considered here. Variations on the order of 5% either way would be typical of similar designs made in several sizes. More substantial variations indicate different evaluations of first cost vs operating costs of the nature already indicated. An example of this would be use of two stages of compression or other complexities usually not encountered in packaged size equipment.

ANNUAL PERFORMANCE

The term Performance Factor is in common use to represent the average COP for the season. This would be expected to have one value for the summer, and another for the winter. Usually the value for the winter is of particular interest. To obtain the performance factor, the input can be measured with electric meters, but the output can only be approximated by estimating the building heat gain or loss, with proper adjustment for internal heat sources such as electric lighting.

The output is subject to the highly complex influences of weather. The output is accordingly subject to large errors, either as a prediction or even as an after-the-fact evaluation.

It would be helpful to have a rule of thumb for predetermining performance factor, but this has never been accomplished successfully. Some of the reasons are already apparent. Other influences should also be mentioned.

Construction features of the building, such as amount of insulation, amount of glass, exposure to the sun, and tightness against air infiltration, are highly important. The building may not actually be built as shown on the plans. Also, corrections must be applied if the entire building is not in use.

The local climate may be different from that given by available weather data. This is true with respect to predictions, but is also true for data from the past. Data may be available only for an airport in the region, which can be quite different from downtown city areas.

Of particular importance is the indoor temperature actually attained. For example, holding a frequently desired indoor temperature of 75 F, instead of the commonly assumed 70 F can increase the heating requirement by one-half in such areas as Dallas, Tex. where the average winter temperature is slightly above 55 F. Dallas has a design temperature of zero. So does Albany, N. Y. However, at Albany the average winter temperature is 35 F, not 55 F. Thus, winter temperature patterns must be carefully established and properly used.

The input to heat pumps, even though metered, must be properly broken down between heating and cooling. Both functions may occur on the same day, and certainly both occur during mild weather between monthly meter readings. Total meter readings for the year do, however, lead to some useful conclusions.

For example, annual electricity usage was between 13,000 and 13,500 kwhr for three 1300 sq ft houses located in south Texas, California and Virginia. This supports the conclusion that the annual power consumption is about the same in different geographic locations, and that as cooling requirements decrease, the heating requirements correspondingly increase.

Another example of four 1700 sq ft houses, one in Alabama and the rest as in the preceding example, in Texas, California and Virginia, showed the same trend wherein total electricity usage for the heat pumps is the same regardless of geographic location. Usage for the four cases was in the narrow range of 11,600 to 13,100 kwhr.

However, in these examples, the 1300 sq ft houses and the 1700 sq ft houses used the same amount of electricity. Certainly this reflects differences in house construction or exposure to sun and wind, rather than geographic location.

In using available data for residences, heating performance factor extremes as high as 4.3 and as low as 1.2 can be calculated. These are not representative performance factors. In general, residential heating season performance factors may be expected from 2.0 to 2.5 and sometimes slightly higher.

In commercial installations, somewhat higher heating season performance factors may be obtained. Lighting loads, body loads and machinery reduce the need for supplementary resistance heating.

A multiple installation in the Grand Central Market in Salt Lake City, Utah demonstrates how the usage of the supplementary heaters can be nullified by the lighting load. Twenty units of 5-ton cooling capacity are located in groups of four on a balcony above the floor area. Each group provides zone control. The supplementary resistance heaters are controlled so that when the lights in a given area are operating, the heaters in the units serving the area are unable to operate. This not only helps reduce the kwhr usage of the system, but it also eliminates possible high billing demands.

In this case the billing demand for the entire market was 20% lower in the winter months than in the summer. The highest monthly electricity use in the winter was 12% less than for the peak month in the summer. Thus heating with heat pumps aided in maintaining a more nearly year-round electric load without introducing a peak load in the winter.

Operating data for a single 5-ton heat pump in a bank in North Carolina are also indicative of how internal sources of heat in a commercial building influence the monthly power requirements for the heat pump. In this case the kwhr requirement in the severest winter month was only 52% of the requirement for the hottest summer month.

A similar 5-ton installation in

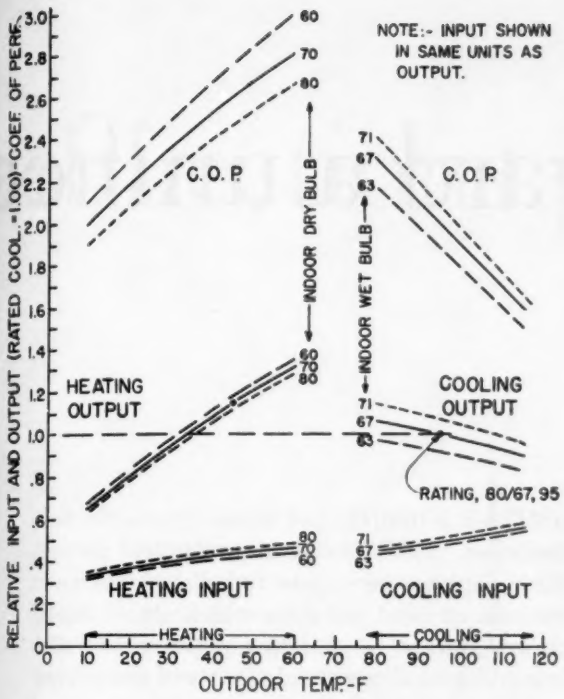


Fig. 4 Typical heat pump performance characteristics (without supplementary heat)

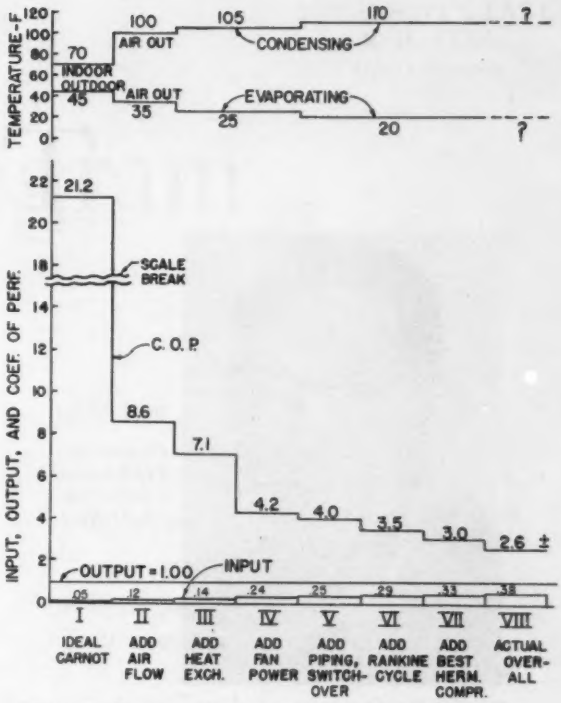


Fig. 5 Temperature pattern and performance analysis

South Carolina in an apparel shop, showed just the reverse of the preceding case. In this case the peak month was in the winter and the highest summer requirement was 51% of that for the peak winter month.

These cases illustrate the substantial effect of internal sources of heat other than the heat pump and explain the suitability of using heat pumps in conjunction with these sources on commercial applications in northern regions. Thus these establishments gain the advantage of heating as well as cooling with little additional investment above that required for cooling only.

Performance characteristics of heat pumps are more complex than conventional methods of heating and cooling and require more analysis and understanding in making applications. Nevertheless, packaged heat pumps are gaining increasing acceptability for year-round air conditioning and are helping utility systems substantially in attaining more nearly uniform summer and winter loads.

WESTERN NEW YORK CHAPTER HOST TO SWEDISH ENGINEERS

Guests of the Western New York Chapter (H) in Mid-February, 26 members of the Swedish Association of Heating and Ventilating Engineers visited Niagara Falls, participated in tours of the nearby American Standard stamping plant and the main works of the Buffalo Forge Company, and inspected the heating, ventilating and air conditioning facilities in the plant of the Buffalo Evening News and the offices of the Bethlehem Steel Company. Hosts and guests are here shown at the brink of the Falls.

Toward a unified pu

R. H. TULL

Chairman
Publications
Committee
ASHRAE

On January 29th, when ASHRAE became a reality, two important engineering societies merged to become a new Society. Each of the predecessor organizations had evolved a Publications Program to best serve its members and the public interest. The publications developed by each had gained acceptance and prestige in the engineering community and a reputation for engineering competency. The publication names such as Refrigerating Engineering, ASHAE Journal, Transactions, the Guide and the ASRE Data Book are widely known and these volumes are to be found on the desks and in the libraries of virtually all who are interested in the related areas of the engineering profession.

Now ASHRAE is developing a new unified Publications Program which must conserve the best values of the programs of the two parent societies. With the greater resources of the combined technical programs, the enlarged research activity and the broader engineering interests of the members to draw from, the new publications should gain greatly in engineering values.

The first of the new publications is the ASHRAE JOURNAL, of which this is the second issue. This monthly magazine will publish those engineering papers and discussions from national or local meetings which are of broad current interest or of lasting value. It will also bring you local news and reports of the overall activities of the Society. It can be expected that the ASHRAE JOURNAL rapidly will become recognized as the foremost engineering publication in the fields of interest of the Society. This is one of the objectives of the Publications Committee.

To serve the members' need for a reference book of fundamental engineering information the ASHAE Guide and the ASRE Data Books have been published annually for a number of years. Each of these publications has a splendid reputation as an authoritative reference volume in its field, but they differ in many ways. For a number of reasons the new Society cannot continue to print both publications. Also, for various reasons, it is not possible immediately to combine the two into a single volume containing high-quality engineering editorial material which has been reviewed, criticized and checked for accuracy and authoritative information. Thus the next volume of the annual reference book will be the 1960 revision of the Guide. This book will be issued to all members of ASHRAE who are entitled to receive it.

Eventually, the Guide and the Data Book must be combined. Because of the broader field of interest of the new Society and the greatly increased volume of engineering information available, it appears certain

PRESIDENT'S PAGE
R. H. TULL
Guest Sponsor

d publications program

that this combination will require the publishing of several volumes. A Guide and Data Book Study Committee is now at work on the problem of determining the content of these volumes and the schedule to be followed in publishing them. It is expected that the first volume of this new series will be published in 1961. Drawing upon the broad engineering knowledge and the experience of the membership of the Society for its content of engineering information, this series should be the foremost, authoritative reference source in all the fields of interest of the Society. This is another objective of the Publications Committee.

The Transactions as published by ASHAE has brought together in an annual volume the engineering information of lasting interest, which had been presented and discussed at the national meetings of that society. This publication also recorded the significant items of business and of historical interest in the year's activities of the society. ASRE had no parallel publication serving these functions. The committee which prepared the merger proposal recognized the value of this publication and recommended that it be continued in the new Society.

It recognized, however, the possibility that only relatively few members preserved and used their copies of Transactions. In view of the considerable cost of publishing this annual volume the committee recommended that the overall policy related to this publication be studied with a view to possible economies. A Transactions Policy Advisory Committee has been appointed as a sub-committee of the Publications Committee to undertake this study.

In order to maintain the continuity in the publication of Transactions, the 1959 volume will be the same in size and format as in the past. It will include engineering papers from the Philadelphia and Lake Placid meetings. It will also include the record of the business meetings which led to and consummated the merger of the two societies as well as the legal documents related to the merger action.

The objective of the Transactions Policy Advisory Committee will be to determine the most suitable way to provide an annual collection of the records of lasting interest to the members at a minimum cost to the Society.

The publications of the Society are intended to provide one of the most valuable, and for many the most tangible, services to the members. Through its publications the Society maintains contact with all members on a uniform, frequent and scheduled program. Also, through its publications as well as in other ways, it attempts to advance the Arts and Science of the profession and raise the level of knowledge of each member. The success of the publications in providing these services can best be judged by the comments received from the members.

These are your publications. If you don't like them tell the editors or the Publications Committee. If you do like them tell us, too. In either case we want your reactions and we solicit your comments and suggestions.

ASHRAE Lab may study atmospheric ions

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Director of Research
ASHRAE Research Laboratory

IONS IN THE ATMOSPHERE

Increasing interest is again being evidenced in regard to ions in the air and their effect on people. Of particular interest at the Philadelphia convention was a presentation which Mr. J. C. Beckett, of San Francisco, Calif., made before two of the Advisory Committees on the Natural Occurrence of Positive and Negative Ions in the Atmosphere. Mr. Beckett had slides showing the various concentration of small and intermediate ions as recorded in several California cities and also in unpopulated mountain regions. He demonstrated the relationship of weather conditions and air pollution to ion concentration and distribution, pointing out that heavy concentrations of positive ions frequently accompany storms and air pollution.

In the discussion following this talk, it was brought out that air-conditioning-system components destroy both positive and negative ions. In particular, high-efficiency air filters remove nearly all ions, both positive and negative. Ion levels in closed rooms parallel outdoor levels but to a lesser extent. On opening windows or providing direct ventilating air, essentially the same ion concentration can be reached indoors as outdoors. The presence of negative ions appears to produce a feeling of well-being in a number of individuals, whereas a high level of positive ions tends to cause uneasiness or discomfort to many individuals. Ions can be artificially created by ionizing apparatus now commercially available.

The whole subject of ionization, in air-conditioning and in the treatment of disease, is well worthy of further investigation. The Laboratory is investigating the possibility of exploratory research in ionization.

RESEARCH ADVISORY COMMITTEE NEWS

Appointment of the Research Advisory Committees is now actively under way, and formal announcement of all the Committees and their membership will be announced soon. The Research Advisory Committees are successor groups to the Technical Advisory Committees of ASHAE, which for so many years effectively served the Research Program and performed many other useful services for the Society. Of significant interest at this time is the newly formed Research Advisory Committee on Refrigeration which will immediately explore directions to be followed and research needs for refrigeration in our Society. It can be announced that Mr. E. P. Palmatier has agreed to serve as Chairman and Mr. D. D. Wile has agreed to serve as Vice Chairman of this Committee.

The Conference on Combustion Research Needs in the Heating Industry has now been definitely set for May 21 at the Research Laboratory in Cleveland. The complete program will be released by the Conference Chairman, Mr. M. W. McRae in mid-April.

DISCOMFORT INDEX

Some comment has recently appeared in the technical press concerning the Discomfort Index (D.I.). This index was developed at the U. S. Weather Bureau in the hope of finding a simple index to express the combined effect of the variables, temperature and humidity. The mathematical expression for the Discomfort Index is:

$$DI = 0.4\,(t_{db} + t_{wb}) + 15$$

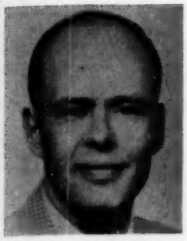
where t_{db} and t_{wb} are the respective dry- and wet-bulb temperatures in degrees F.

A value of D.I. (Discomfort Index) of 70 F is considered the critical value, and whenever values of the index are above 70 F for summer conditions, discomfort may exist.

The optimum line for summer on the ASHAE comfort chart coincides with 71 F effective temperature, and along this line corresponding values of the Discomfort Index range from 70 to 71.8 F. There is consequently no disagreement with the general premise that, as D.I. values rise appreciably above 70 F, discomfort can be anticipated. However, Discomfort Index appears to be little more than a simple formulation relating the comfort of the individual to an outdoor environment.

Discomfort Index may thus have utility in expressing weather information, but as far as air conditioning is concerned, it contributes nothing new to information which has been at our disposal for many years. The quite unfortunate thing is that the Discomfort Index can be extremely misleading to uninformed individuals who, from an analysis of dry-bulb and wet-bulb temperatures, might gain the impression that no discomfort problems arise if outside temperatures are such that the D.I. values remain below 70 F. Unfortunately, with humans living in buildings, nothing could be further from the truth, as the only people in a building whose comfort would even be approximated by this formula would be those who spend their time on building patios, in penthouses, or in the peripheral rooms of a building with no human, lighting or solar loads and with all windows open. These conditions are rarely reached.

Viscosity-solubility characteristics of mixtures of Refrigerant-13B1 and lubricating oils



LYLE F. ALBRIGHT

JAMES D. LAWYER

Halogenated hydrocarbons (primarily chloro and chlorofluoro derivatives of methane, ethane, and propane) have found widespread use as refrigerants. Bromotrifluoromethane ($CBrF_3$) has a critical temperature of 152.6 F (5) and a normal boiling point of —71.95 F (5) and has recently been suggested as a low-temperature refrigerant, Refrigerant-13B1.

Halogenated hydrocarbons have a rather high solubility in lubricating oils, and the resulting oil-refrigerant mixture has a viscosity significantly lower than that of the original oil. Information on the solubility and viscosity characteristics of the refrigerant and oil is hence of considerable value in designing the refrigerant compressor. The viscosity and solubility characteristics have been reported (1, 2, 4, 8, 9) for refrigerant-oil mixtures containing CCl_3F (Refrigerant-11), CCl_2F_2 (Refrigerant-12), $CClF_3$ (Refrigerant-13), $CHClF_2$ (Refrigerant-22) and C_2ClF_5 (Refrigerant-115). In all cases the oil absorbed a considerable amount of the halogenated refrigerant and the viscosity was reduced appreciably by the absorption of small quantities of the refrigerant. The solubility of the refrigerant increased with an increase in pressure or with a decrease in temperature. The lubricating oils were described sufficiently in only three cases (1, 2, 4) in order to compare the results. In general, paraffinic-base oils absorb about 10% more refrigerant on a mole basis than naphthenic-base oils, but on a weight basis this trend is reversed. The viscosity of naphthenic-base oils is reduced more than paraffinic-base oils on both a comparable weight and mole basis.

An investigation was made to determine the solubility and viscosity characteristics of several mixtures of bromotrifluoromethane and representative lubricating oils. Data were obtained over a temperature range of 50 to 150F and to pressures approaching the vapor pressure of Refrigerant-13B1.

Apparatus and Materials. The solubility and viscosity apparatus used had essentially the same design as that used by Albright and Mandelbaum (1). The operating and calculating procedures used were also similar. The only exception was that in determining the amount of Refrigerant-13B1 dissolved in the oil, the volume of the liquid phase was assumed to increase because of the dissolved refrigerant.

Subsequently, the new liquid volume was calculated assuming the volumes of the oil and refrigerant were additive, which is true for ideal solutions. Martin (5) has recently reported a reliable equation of state for bromotrifluoromethane and it was used to calculate the amounts of Refrigerant-13B1 in the gas state. E. I. du Pont de Nemours and Co., Inc. reported that the only detectable impurities in the Refrigerant-13B1 used were 0.07 mole per cent CCl_2F_2, 0.3 mole per cent $CHBrF_2$, and 0.01 mole per cent C_2BrF_5. The water content was indicated to be less than 15 ppm. The characteristics of the four lubri-

TABLE I

IDENTIFICATION AND CHARACTERIZING TESTS OF OILS

Identification:	80 SUS*	300 SUS*	150 SUS**	300 SUS**
Tests:				
Gravity — API	27.4	23.8	31.6	29.0
Flash, O-Cleve F	320	405	410	455
Fire, Cleve F	360	455	470	520
Viscosity, Kinematic CS at 100 F	16.38	67.11	31.96	72.72
Viscosity, Kinematic CS at 210 F	3.16	6.66	5.14	8.35
Viscosity, Saybolt Univ. Sec. at 100 F	82.7	310	149.5	336
Viscosity, Saybolt Univ. Sec. at 210 F	36.8	47.9	43.1	53.6
VI (Viscosity, Index)	37.2	31	97	91
Molecular Weight	310	380	400	440
Density, g/cc				
50 F	0.894	0.914	0.871	0.884
75 F	0.885	0.906	0.862	0.876
100 F	0.876	0.897	0.854	0.867
125 F	0.867	0.889	0.845	0.859
150 F	0.859	0.882	0.836	0.855

* Solvent refined naphthenic distillate.
** Solvent refined paraffinic distillate.

Lyle F. Albright is Professor, Chemical Engineering Department, Purdue University. James D. Lawyer is with the University of Oklahoma.
This paper was presented as "Viscosity and Solubility Characteristics of Mixtures of Refrigerant—13B1 and Lubricating Oils" at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958.

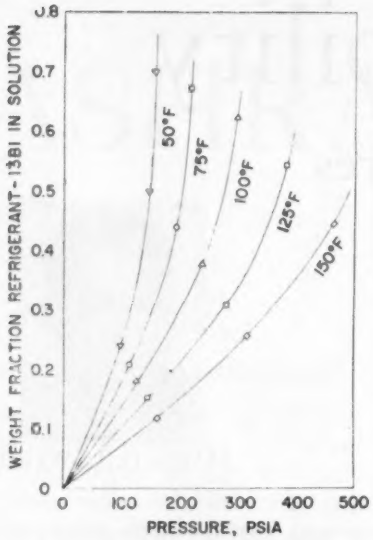


Fig. 1 Solubility of Refrigerant-13B1 in 80 SUS Naphthenic Oil

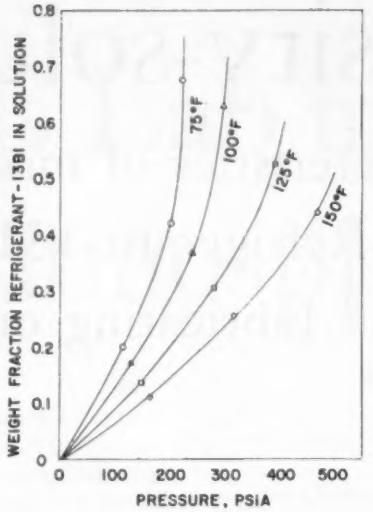


Fig. 2 Solubility of Refrigerant-13B1 in 300 SUS Naphthenic Oil

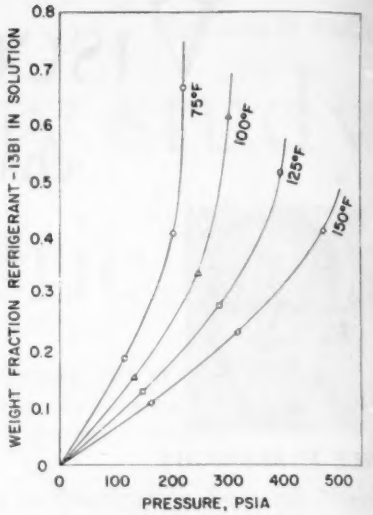


Fig. 3 Solubility of Refrigerant-13B1 in 150 SUS Paraffinic Oil

cating oils furnished by the Texas Company are given in Table I. These four oils had been used previously in a similar investigation by Albright and Mandelbaum (1).

Discussion of Solubility Results. Solubility data were obtained for three representative lubricating oils and Refrigerant-13B1. The experimental solubility data on a weight basis are presented in Figs. 1, 2, and 3, in which the solubility expressed as weight per cent of Refrigerant-13B1 in the solution is plotted vs. the absolute refrigerant pressure with temperature as parameter. Pressures studied were up to about the vapor pressure of Refrigerant-13B1 over a temperature range of 75 to 150 F or 50 to 150F. The solubility of Refrigerant-13B1 in the oils increased in all cases as the pressure increased and approached a composition of a 100% refrigerant as the pressure approached the vapor pressure of Refrigerant-13B1. At lower temperatures it was also found that more refrigerant was dissolved.

Figs. 1 and 2 indicate that on a weight basis the low-viscosity naphthenic oil absorbed more refrigerant at the same temperature and pressure than the high-viscosity oil. The same trend was reported by Albright and Mandelbaum (1) for Refrigerants-13 and -115 and by Little (3) for Refrigerant-22. The solubility results were converted to a mole basis (see Fig. 4) using molecular weights of the oils estimated by the method of Hougen and Watson (3) who indicated that the molecular weights determined by their correlations are generally correct within 5%. On a mole basis, the solubilities of Refrigerant-13B1 in the two naphthenic oils of relatively different viscosities were found to be essentially identical. Albright and Mandelbaum had previously reported a similar finding for Refrigerants-13 and -115 in both naphthenic and paraffinic-base oils. On a mole basis, however, Refrigerant-13B1 is about 10% more soluble in paraffinic oils than in naphthenic oils. On the basis of the present results and those of the previous investigation (1), it can be assumed that relatively accurate generalized plots of the solubility on a mole basis can be determined for both naphthenic and paraffinic oils.

The reliability of the solubility data obtained in this investigation depended upon the accuracy in determining the density of the oil, the volume of the oil chamber, the volume of the refrigerant chamber, the temperature and pressure of the system, the equation of state used for Refrigerant-13B1 and the volume for the oil-refrigerant solution. Consideration of the various possible inaccuracies indicates the probable maximum error for the solubility data to be approx. 4% in the high solubility range and about 7% in the low solubility range. The largest error probably involved the assumption that the volume of the oil-refrigerant mixture was equal to the volume of the original oil plus the volume of the dissolved refrigerant liquid. In both the solubility and the viscosity measurement, some error may have been introduced by the fact that equilibrium was not reached. Because of the precautions used equilibrium was probably reached in all cases. Such an assumption seems likely because of several reruns of the data and good correlation of the data.

Othmer and White (6) have presented a method of correlating the solubility of gases in liquids by means of the following equation:

$$\log P = A \log p + B \qquad (1)$$

The equation was applied to the solubility data of this study. The vapor pressure, p, used was that of the refrigerant. A graph of the solubility results for mixtures of Refrigerant-13B1 and naphthenic oils is shown in Fig. 5. A similar plot was obtained for the mixture containing the paraffinic base oil. The abscissa of Fig. 5 is in temperature units, which were obtained from the vapor pressure-temperature relationship of the refrigerant. Straight line correlations were obtained for constant compositions as the equation predicts. When the ratio of the heat of solution to the heat of vaporization is constant, the equation presumably will be valid, i.e., straight line correlations will be obtained (6). Such an assumption is probably quite accurate over a temperature range extending somewhat beyond 50 to 150F.

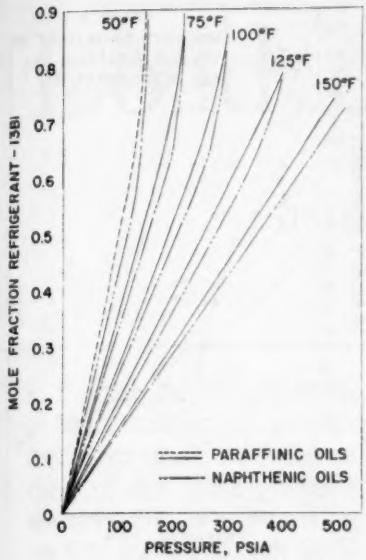


Fig. 4 Solubility of Refrigerant-13B1 in Naphthenic and Paraffinic Oils

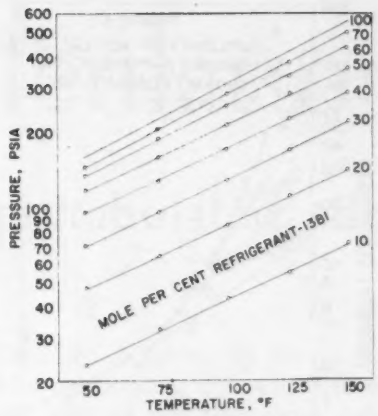


Fig. 5 Solubility of Refrigerant-13B1 in Naphthenic Oils

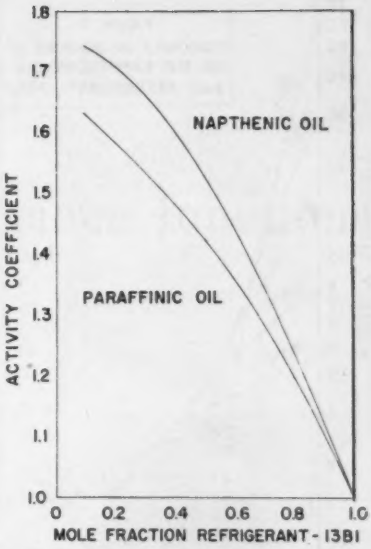


Fig. 6 Activity coefficients of Refrigerant-13B1 in Naphthenic and Paraffinic Oils

In the case of the system containing Refrigerant-13B1 and the paraffinic base oil, the straight lines were extrapolated to 50F, in order to determine the 50F curve on Fig. 4. Equation 1 was also found to be applicable for the solubility data of Albright and Mandelbaum (1) for Refrigerants-13, -22, and -115. Presumably Equation 1 is applicable to other oil-refrigerant systems in at least the 50 to 150F range.

The activity coefficients of Refrigerant-13B1 were calculated for the oil-refrigerant mixtures by the method outlined by Albright and Mandelbaum (1). The values of the coefficients were found to decrease toward 1.0 with increased refrigerant concentration as shown in Fig. 6. At a refrigerant composition of 5 mole per cent, the activity coefficient was about 1.7. Within experimental accuracy the activity coefficients were not changed by temperature in the range of temperature studied; however, the values for paraffinic oils are slightly less than those of naphthenic oils. The activity coefficients for Refrigerant-13B1 are considerably lower than those reported (1) for Refrigerants-13 and -115. The values are similar to those for Refrigerant-22, and higher than those of Refrigerant-12. Since activity coefficients are quantitative measurements of the ideality of solutions, the solutions containing Refrigerant-13B1 are more ideal than those containing Refrigerants-13 and -115, and hence there is less possibility of forming immiscible liquid layers. It is quite probable that the solutions studied in this investigation were completely miscible over the entire range of temperatures and pressures studied.

Albright and Mandelbaum (1) reported previously that the replacement of a chlorine atom with a fluorine atom in the refrigerant molecule causes less ideal solutions of the refrigerant and oil. Comparison of the present results with the results for mixtures of oil and Refrigerant-13 (1) indicates that the replacement of a chlorine atom with a bromine atom in the refrigerant results in more ideal oil-refrigerant mixtures. Apparently the higher the molecular weight of the halogen substituted in the refrigerant the more ideal the oil-refrigerant mixture obtained.

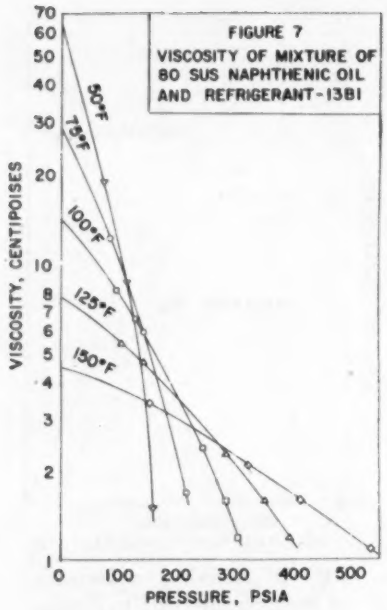
Discussion of Viscosity Results. The viscosity of mixtures of Refrigerant-13B1 and four representative lubricating oils were obtained in the present investigation, and are shown graphically in Figs. 7, 8, 9 and 10. In these graphs the logarithm of viscosity was plotted vs. absolute pressure with temperature being parameter. An attempt was made to evaluate the experimental inaccuracies of the viscosity measurements such as the time of roll of the ball, the density of the viscometer ball, the density of the fluid under consideration, the temperature and pressure of the system, and the methods of calibrating the viscometer. In calculating the viscosity of the oil-refrigerant mixtures, the density of the mixture was assumed to be the same as that of the original oil. Such an assumption will of course cause slight errors in the data which were calculated to be in the order of 1% or equivalent to about 0.2 cp. The total errors of the viscosity data are thought to be less than 1.0 cp in all cases.

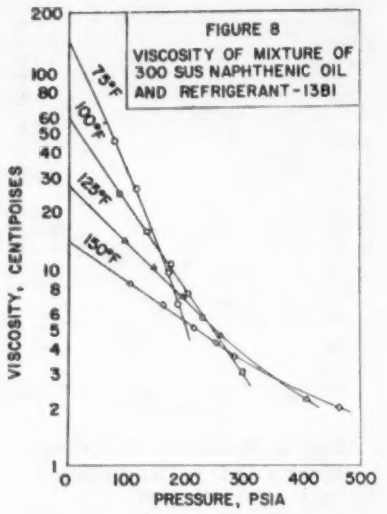
On both a weight and mole basis Refrigerant-13B1 reduces the viscosity of naphthenic oils to a greater extent than paraffinic oils. The same trend was reported earlier by Albright and Mandelbaum (1). The viscosities of the solutions of Refrigerant-13B1 and oils were found to be appreciably lower than those of the pure oil. At pressures approaching the vapor pressure of the refrigerant, the viscosity was reduced to values of 1 or 2 cp.

Othmer and Conwell (7) have reported that the viscosity of liquids may be correlated by the following equation:

$$\log n = -D \log p + F \qquad (2)$$

This equation was found to correlate successfully the data of this investigation and those of Albright and Mandelbaum (1) for Refrigerants-13 and -115 in oil-refrigerant mixtures. Fig. 11 for the 150 SUS paraffinic oil mixture demonstrates a typical correlation obtained. The straight lines obtained for a given composition can probably be extrapolated to higher and lower temperatures.





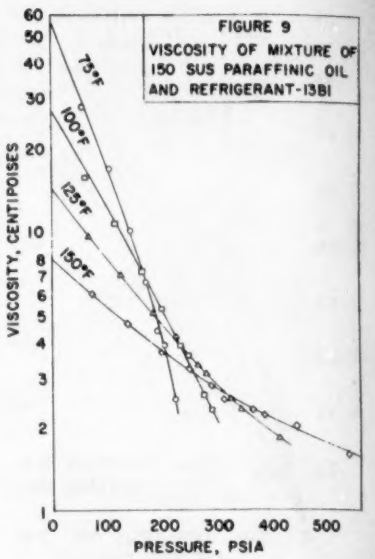


Fig. 12 shows the viscosity reduction of a 150 SUS paraffinic base oil with Refrigerant-22, Refrigerant-13, Refrigerant-115 and Refrigerant-13B1. This plot indicates that Refrigerant-13B1 reduces the viscosity of the lubricating oil about the same as the other refrigerants when the viscosity is plotted vs. the mole fraction of refrigerant in the solution. Such a plot indicates that the reduction in viscosity of the oil depends primarily upon the amount of refrigerant in the oil and is relatively independent of the particular refrigerant used.

CONCLUSIONS

The solubility of gaseous Refrigerant-13B1 increased in four typical lubricating oils as the pressure increased and as the temperature decreased. At pressures approaching the vapor pressure of the refrigerant, the composition of the oil-refrigerant mixture approaches 100% refrigerant. Increased amounts of dissolved Refrigerant-13B1 in the oils significantly decreased the viscosity of the mixture, to as low as 1 to 2 cp. Both the solubility and viscosity data were correlated by methods presented by Othmer and co-workers. These correlations apparently can be extrapolated with reasonably good accuracy and they will be of value for designing refrigeration systems using Refrigerant-13B1.

NOMENCLATURE

A Constant in Equation 1
B Constant in Equation 1
D Constant in Equation 2
F Constant in Equation 2
n viscosity, cp
p vapor pressure of refrigerant
P absolute pressure of mixture at equilibrium

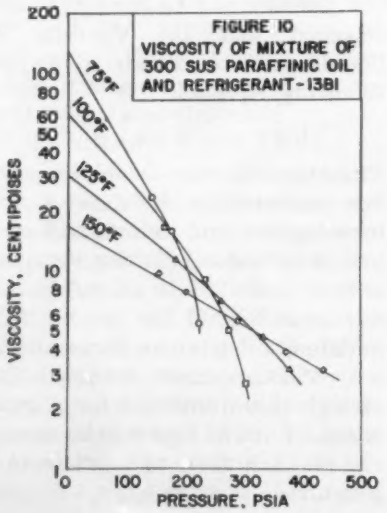
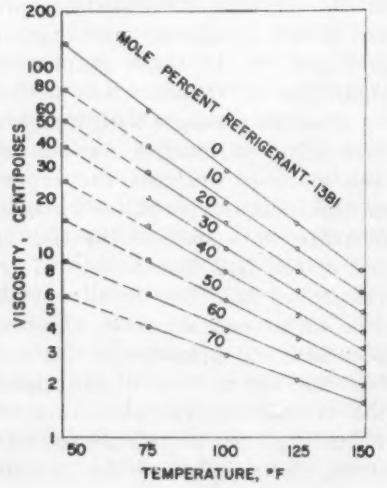


Fig. 11 Viscosity of mixture of 150 SUS Paraffinic Oil and Refrigerant-13B1



ACKNOWLEDGMENT

The Texas Company furnished the lubricating oils used and the E. I. du Pont Company furnished the bromotrifluoromethane.

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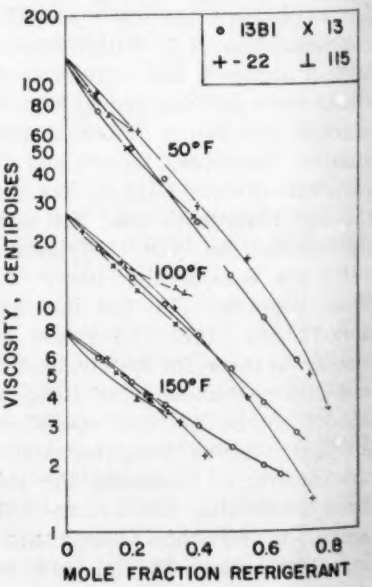
Fig. 12 Viscosity of mixtures of various refrigerants and 150 SUS Paraffinic Oil



Load calculations

using pretabulated admittance functions

As a simple and accurate method of load calculation this procedure requires no special computing machines. It does utilize information obtained previously with computing machines and is sufficiently flexible to allow the use of local time variable weather data. The results of calculations are compared with those based on equivalent temperature differentials given in The ASHAE Guide of 1957.

Before the widespread use of modern computers, Mackey and Wright[1,2,3] developed methods for the computation of periodic heat transfer at the inside surface of separate homogeneous wall sections. These computational procedures are based on the analytical solution of the differential equation of heat conduction and involve certain approximations. Johnson[4] extended this work presenting temperature amplitude decrement and time lag values in terms of generalized property and boundary conductance moduli for homogeneous walls.

Equivalent temperature differentials for calculating heat gains through structural sections are tabulated in The Guide. These data are based on the Mackey and Wright solutions and computed for a fixed value of surface conductance and surface absorptivity. Stewart,[5] in setting up the equivalent temperature differential method, pointed out that some judgments and arbitrary adjustments were made as a necessary expedient. In addition, the temperature differentials are tied to single sol-air temperature functions and corrections that are not adequate for many applications.

Greater economy and precision in engineering analysis can now be achieved through the use of machine calculations. The approach to be taken in any particular application depends upon the objectives, the availability of computers, and the qualifications of personnel. In this study a rational approach is made to the use of machine computed data that can be tabulated in advance for the determination of heat fluxes through structural sections. The form of the data tabulated is such as to make it independent of the boundary conditions and directly applicable to any situation. Limitations of the calculation procedure to be described are discussed in another section.

Beginning with the thermal network representation of an enclosed space, each conduction path network may be represented by a transfer impedance function (t_{in}/q) as shown in Reference 6. The transfer impedance thus defined is a function of input frequency and boundary resistance as well as thermal properties of the structure itself. In Reference 6 it was suggested that the transfer function be expressed in terms of the characteristic thermal impedance offered by a given construction and the separate boundary resistance with the view of pretabulating characteristic impedances for typical constructions. It now appears more desirable to precompute the transfer admittance function q/t_{in} of typical constructions for various values of boundary resistance. This eliminates the necessity of solving the complex algebraic expression for t_{in}/q when making load calculations. Further, it has been demonstrated[7] that transfer functions can be computed and tabulated so economically and rapidly with an ac electric network analyzer that it would be a relatively small task to precompute and tabulate the transfer admittance functions over a wide range of boundary resistances for all types of construction for which thermal properties are known.

The methods used in precomputing transfer functions with an ac network analyzer and an investigation of the influence of lumping, the importance of sol-air temperature harmonics, the importance of accurately determining the convective conductances, and the effects of absorbed radiation on inside surface temperatures are presented in a companion paper listed as Reference 7.

W. B. DRAKE

HARRY BUCHBERG
D. LEBELL

METHOD FOR LOAD PREDICTION

Many factors are involved in sensible load calculations including: infiltration; ventilation; heat gain through walls, ceiling, floor and glass areas; inside radiation exchange and internal heat sources.

W. B. Drake is Research Engineer, Missiles and Space Div., Lockheed Aircraft Corporation. Harry Buchberg is Associate Professor of Engineering, Dept of Engineering, University of California. D. Lebell is a Consulting Engineer and Part-time Lecturer, Dept of Engineering, University of California.

This paper is based on one part of an investigation carried out in partial satisfaction of the requirements for the degree of Master of Science in Engineering. It is the result of research sponsored by the American Society of Heating and Air-Conditioning Engineers in cooperation with the Dept of Engineering, University of California and is planned for presentation at an early National Meeting of ASHRAE.

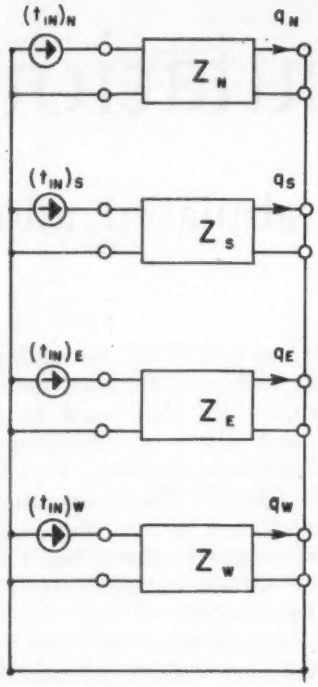
a. PLAN VIEW

Fig. 1a Single room enclosure – plan view

The method of calculation presented here considers heat gain through walls only, but it can be extended to include other heat flow paths by the method of superposition. It should be recognized that the method presented is a simplified steady-state sinusoidal solution utilizing a maximum amount of pretabulated information.

The suggested method for load prediction will be explained through the use of an example problem which considers a single enclosure, shown in Fig. 1a. The thermal circuit to be solved is shown in Fig. 1b. A cross-section of the wall structure is shown as Fig. 1c in the example calculation. The problem is solved by determining the load contribution of each wall separately, based on a constant and uniformly distributed inside air temperature.

To calculate the heat flow through individual wall sections, the designer will need: (1) a knowledge of the wall construction and appropriate transfer functions; (2) the inside design temperature; and (3) the input potential expressed as sinusoidal components of a sol-air temperature application to the various orientations of the structure under design. Knowledge of these three quantities will enable the designer easily and simply to construct a graph showing the variation in cooling load with time for a 24-hr period. Values of steady-state resistance and transfer functions computed in Reference 7 for several types of construction are given in Table I. The transfer admittance functions are represented by the mean-to-peak magnitude M, of heat flux per unit area at the inside boundary resulting from a mean-to-peak magnitude of 1 deg temperature variation applied at the outside boundary. The phase angle, β, in hours is the time lag between the resulting heat flux and



b. CIRCUIT

Fig. 1b Single room enclosure – circuit

applied temperature input. Values of 4.0 Btu/hr/sq ft/F for the outside boundary conductance (h_o) and 1.65 Btu/hr/sq ft/F for inside boundary conductance h_i were used in the computation of steady state resistances and transfer admittance functions.

Sol-air temperatures given in Table 8, p. 302 of The Guide of 1957 were used for the sample calculation in order to compare results with calculations based on equivalent temperature differentials. The data given in Table II were obtained by a Fourier analysis of the sol-air temperatures following the procedure suggested.[8]

The procedure for determining the heat flux for a single wall is as follows with the south and west facing walls being used in this example.

For a given wall construction such as 8-in. common brick (Fig. 1c) obtain values for R_t, M, and β from Table I: R_t = 3.73 F/hr/sq ft/Btu; 1st harmonic Frequency, M = 0.121 Btu/hr/sq ft/F; β = 6.8 hr. At 2nd harmonic frequency, M = 0.0216 Btu/hr/sq ft/F; β = 4.8 hr.

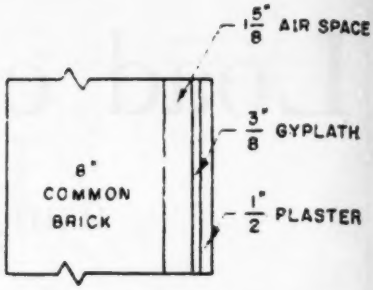


Fig. 1c Cross section of wall construction in example

Determine the wall area, A for each orientation. For this example calculations are made on the basis of 1 sq ft wall area.

Decide on the inside design temperature; let t_i = 80 F.

Obtain the necessary sol-air temperature data. From Table II, ${}_mt_{sa}$ = 89 for south and for 1st harmonic ${}_{mp}t_{sa}$ is 18 F, while θ_{sa} is 13.8 hr. Similarly, ${}_mt_{sa}$ = 93 for west and for 1st harmonic ${}_{mp}t_{sa}$ is 27.8 F while θ_{sa} is 15.8 hr.

Calculate the constant value or dc component of heat flux, $q_{dc} = [({}_mt_{sa}) - t_i]/R_t$, and obtain 2.41 and 3.48 Btu/hr/sq ft South and West respectively.

Calculate the maximum value of heat flux, $q_{max} = q_{dc} + M ({}_{mp}t_{sa})$, and obtain 4.59 and 6.84 Btu/hr/sq ft for South and West respectively.

Calculate the minimum value of heat flux, $q_{min} = q_{dc} - M ({}_{mp}t_{sa})$, and obtain 0.23 and 0.12 Btu/hr/sq ft for South and West respectively.

Determine the time at which the peak value of heat flux occurs. This is found by adding θ_{sa} and β giving 20.6 hr past midnight for the south wall and 22.6 hr past midnight for the west wall.

Compute the daily variation in heat flux for each wall. The contribution of each wall to the total load may now be determined by using Fig. 2 as a preprinted curve and labeling the axes in accordance with the computation of q_{dc}, q_{max}, q_{min} and the values of β

and θ_{sa}. It should be recognized that a sinusoidal excitation or input always produces a sinusoidal response resulting in a quite simple determination of the response when the input and transfer function is known. Thus, for the south wall example, the values for q_{dc}, q_{max}, and q_{min} are entered on the ordinate of Fig. 2. This is shown in Fig. 3 with the intermediate values added. The time scale is established by placing 20.6 hr at the peak heat flux and marking off the entire scale in hours on both sides of the peak. Sinusoids for additional input harmonics may be determined in the same manner. The analytical expressions for the south and west wall heat fluxes can also be easily determined. Based on 2 input harmonics, they are

$$q_s = 2.41 + 2.18 \cos [15\,\theta + 51^\circ] + 0.148 \cos [30\,\theta - 83^\circ]$$

$$q_w = 3.48 + 3.24 \cos [15\,\theta + 18^\circ] + 0.294 \cos [30\,\theta - 185^\circ]$$

Determine the daily variation in heat load. The total load usually includes contributions due to transmission, solar input, ventilation, and various heat sources such as lights, occupants, and solar power transmitted through glass areas. Curves giving the daily variation in heat flux including solar effects are constructed for each of the walls as shown in the example just given. The ventilation load curve is determined in the usual manner (infiltration effects may be added to this curve). Constant and sinusoidally varying radiation sources may be treated as shown in Reference 2, section on Radiation Sources. The final step in the computation is to add the values of q, hour by hour, to obtain the final load curve.

DISCUSSION

Certain limitations in the steady state sinusoidal method of load calculation appear at present. As outlined, the method assumes a constant indoor design temperature. Thus, the designer cannot account for the capacitive effects of interior partitions, furnishings, and materials stored in the space. Likewise, the effects of control settings and various types of thermostats cannot easily be studied. In cases where transients are important, the steady state sinusoidal approach has definite limitations. Under these circumstances the transient response method[9] of network solution or dc network analyzer solutions[10] are the most promising.

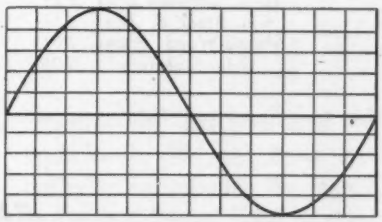


Fig. 2 Blank graph for use in plotting results of calculations

In the specification of cooling systems, the complexity of additional harmonics is not necessary in many instances. Design conditions, particularly sol-air temperature inputs, are arrived at through a combination of experience and weighted weather data. In building design, wall heat fluxes do not usually represent a large portion of the sensible load. Because of these facts it is believed that a sinusoid can be obtained which would represent a specified design condition and result in reasonably accurate load calculations. However, the load contributed by additional input harmonics can easily be computed when necessary. For example, instead of a single graph for each wall, as shown in Fig. 3, it would be necessary to prepare an additional graph for each additional harmonic based on a knowledge of the transfer functions for each additional frequency.

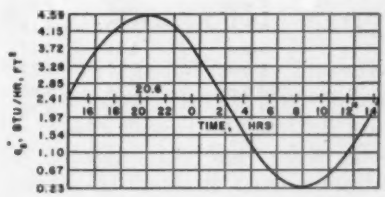


Fig. 3 Graph with sample values plotted. Daily heat flow through south facing 8-in. common brick wall

Radiation exchange between interior surfaces can be included in load calculations through a reiterative procedure as shown in Reference 11. A less cumbersome procedure is the insertion of a resistance network representing radiation exchange and the solution by direct electric analogue techniques given in Reference 10. However, these methods do not lend themselves easily to the simplified hand procedure described in this paper. For the case of constant and sinusoidally varying radiation sources, tables such as Tables 6 and 7 in Reference 7 make it convenient to include the resulting load and to determine the temperature rise of the receiving surfaces.

By inspecting Table III one may compare the values of heat flux computed by the suggested method with values based on equivalent temperature differentials given in The Guide of 1958 for the same construction and sol-air inputs. A U value of 0.27 was used for the brick wall section consistent with resistances given on pp. 177 and 186 of this edition of The Guide. It will be noted that the peak heat flux predicted by the suggested method using two input harmonics is less than 3 percent greater than the equivalent temperature differential method but values at other times differ much more. It would be rather difficult to determine precisely the reasons for these differences in view of the fact that certain approximations and arbitrary adjustments were made in computing the equivalent temperature differentials.[5] From a practical point of view, the difference in peak values predicted by the two methods is insignificant.

The main attributes of the suggested method of load calculation presented in this paper are the following ones:

Simplicity: Requires only a small amount of calculation utilizing previously machine computed local input data and transfer functions of typical constructions. A simple graphical procedure using preprinted sinusoid curves on a grid may be used to display hourly

values and the final load curve determined by summation of hourly data.

Flexibility: Can accommodate local variations in weather through the use of the appropriate sol-air input and surface conductances.

Accuracy: Obtained within the limitations discussed previously. The number of input harmonics required in a given instance depends upon how much of the total load is contributed by the wall heat fluxes, the shape of the sol-air input curve, and the type of construction. Any number of harmonics can easily be handled with this method once the transfer functions have been computed. In many instances a single harmonic is sufficient for equipment sizing purposes. A formula for the number of harmonics to be used in any application is not possible.

Additional refinement in load analysis requires an improvement in the knowledge of the thermal properties of materials and composite wall constructions, of microclimate effects on boundary inputs and resistances, and in the incorporation of more of the influencing factors into the thermal network.

TABLE I

Tabulated transfer functions of wall sections for use in calculation of cooling loads by steady state sinusoidal method

Wall Section, with $h_o = 4.0$, $h_i = 1.65$	R_t	M		β	
8-in. Common Brick Insulated	8.28	0.048		9.91	
		1st Harmonic	2nd Harmonic	1st Harmonic	2nd Harmonic
8-in. Common Brick	3.73	0.121	0.0216	6.8	4.8
12-in. Common Brick	3.39	0.0865		9.11	
4-in. Brock 6" Tile	3.37	0.154		6.86	
Stucco-Wood Frame	2.07	0.460		0.94	
10-in. Concrete	5.01	0.0747		7.52	

TABLE II

Sol-air temperature data
(Obtained by Fourier Analysis following procedure given in Reference 8)

Orientation	$_{mpt_{sa}}$	$_{mpt_{sa}}$ 1st harmonic	$_{mpt_{sa}}$ 2nd harmonic	θ_{sa} 1st harmonic	θ_{sa} 2nd harmonic
South	89	18.0	6.84	13.8	0.73
West	93	27.8	13.6	15.8	7.5

TABLE III

Comparison between wall heat fluxes based on equivalent temperature differentials given in The Guide and new method

	Heat Flux, Btu per (hr) (sq ft) South Wall			West Wall		
		New Method			New Method	
	1958 Guide	One Harmonic	Two Harmonics	1958 Guide	One Harmonic	Two Harmonics
8 a.m.	1.08	0.25	0.11	2.16	0.90	1.07
10 a.m.	1.08	0.37	0.25	1.62	0.17	0.05
12 a.m.	1.08	1.04	1.06	1.62	0.32	0.03
2 p.m.	1.08	2.07	2.21	2.16	1.32	1.15
4 p.m.	2.70	3.19	3.31	2.70	2.90	3.02
6 pm	4.31	4.11	4.09	3.78	4.63	4.92
8 p.m.	4.31	4.57	4.43	5.40	6.06	6.23
10 p.m.	3.24	4.45	4.33	6.48	6.79	6.67
12 p.m.	2.70	3.78	3.80	6.48	6.54	6.25

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NOMENCLATURE

A = heat transfer area, square feet.
h_i = inside boundary conductance, Btu per (hour) (square foot) (Fahrenheit degrees).
h_o = outside boundary conductance, Btu per (hour) (square foot) (Fahrenheit degrees).
M = mean-to-peak magnitude of heat flux per unit area at inside boundary per unit temperature difference, $[_{mpt_{sa}} - t_i]$ = Btu per (hour) (square foot) (Fahrenheit degrees).
q = heat flux at inside boundary, Btu per hour.
R = resistance to heat transfer, Fahrenheit degrees per (Btu per hour) (square foot).
R_t = total steady state heat transfer resistance, Fahrenheit degrees per (Btu per hour) (square foot.)
t_i = inside design temperature, Fahrenheit degrees.
t_{ia} = time-variable outside design potential or sol-air temperature, Fahrenheit degrees.
$_{mt_{sa}}$ = mean value of sol-air temperature, Fahrenheit degrees.
$_{mpt_{sa}}$ = mean-to-peak value of sol-air temperature (1st harmonic), Fahrenheit degrees.
z = transfer function (t_{ia}/q), complex quantity represented by M and β.
β = phase angle or time lag between resulting heat flux (q) and applied temperature (t_{ia}), hours.
θ = hours past midnight.
θ_{sa} = time of occurrence of peak value of sol-air temperature, hours past midnight.
E = east wall.
N = north wall.
S = south wall.
W = west wall.
max = maximum value.
min = minimum value.

Polyester film insulation

for hermetic motors

J. P. HARRINGTON

Polyester film* has been used as insulation in the open motor industry for approximately five years. In this short interval, it has become a leading insulating medium. Its use is now being extended to ground and phase insulation in hermetic motors with much success.

R. J. WARD

In the use of polyester film as hermetic motor insulation, there are two phases. First, its advantages in design and production and, secondly, its performance in an hermetic refrigeration environment.

In motor design, polyester film offers definite advantages over previously used hermetic insulation. Size reduction of the motor and the possibility of higher operating temperatures are two of the advantages gained when using this new, thin insulating material.

In production, polyester film works extremely well on existing automatic equipment. On slot liner insertion machines its high tensile strength practically eliminates the insulation breakage that occurs with rag paper.

Cold forming of slot liners from polyester film can usually be done with dies previously designed for paper, although small die changes are sometimes necessary. In some wedge applications it may be necessary to heat form the film (at about 340 F) in order to get the desired configuration.

On the basis of detailed laboratory evaluation confirmed by several motor manufacturers in actual production, polyester film was found to have a cut-through resistance superior to all commonly used insulating materials on an equivalent thickness basis. Evaluation was made on a pendulum type impact tester simulating the forces developed by automatic coil winding equipment.

IN HERMETIC REFRIGERATION

In the evaluation of any insulation for use in an hermetic environment there are three things which must be considered. First, what effect the insulation will have on other materials in the system; second, what effect the insulation will have on the operation of the system; and third, what effect the system will have on the insulation.

With these three points in mind let us first take a look at the effect polyester film will have on other materials in an hermetic system.

Sealed tube tests show that this film has no significant effect on the stability of the various materials found in hermetic systems. In a series of tests run for six months at 300 F. there was no indication that its presence accelerated corrosion of the metal strips or "plating out" of copper on the steel strips exposed to Refrigerant-22** and oil. Similar results were obtained with both Refrigerant-22 and Refrigerant-12 in tests run for eight months at 250 F, even when water was added to the sealed tubes. Analysis for chloride ion content in all these tests showed no adverse effect on the decomposition rate of the refrigerant (see Table I).

The second point is the effect of polyester film on the operation of the hermetic system. It has been established that this film contains a small percentage of low molecular weight materials which can be extracted by some solvents at elevated temperatures, e.g. 1 to 1.5% extractables have been removed by boiling in xylene at 285 F. However, sealed tube tests indicate that these extractables are fine particles and are not of a gummy, cohesive nature. Therefore, they should not cause capillary plugging. We do not know of a single instance where reciprocating type hermetic systems containing this polyester film have failed due to capillary plugging, bearing contamination, or similar conditions.

The low moisture absorption and exceptional dimensional stabil-

J. P. Harrington is a technical representative, R. J. Ward is a technical specialist, both with the Chestnut Run Laboratory, Film Dept, E. I. duPont de Nemours & Co. This paper was presented as "Hermetic Motors Insulated with Polyester Film" at the Domestic Refrigerator Engineering Conference at the 45th Semi-annual Meeting of ASRE, New Orleans, La, December 1-3, 1958.

*Produced from polyethylene terephthalate by E. I. duPont de Nemours & Co. as "Mylar".

**duPont has the registered trademark of "Freon 22" for this fluorinated hydrocarbon of its manufacture.

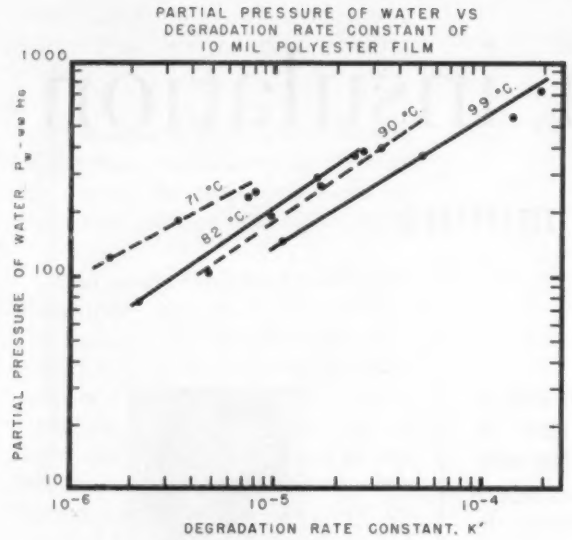


Fig. 1 Comparison of partial pressure of water with the degradation rate constant at specific temperatures and humidities

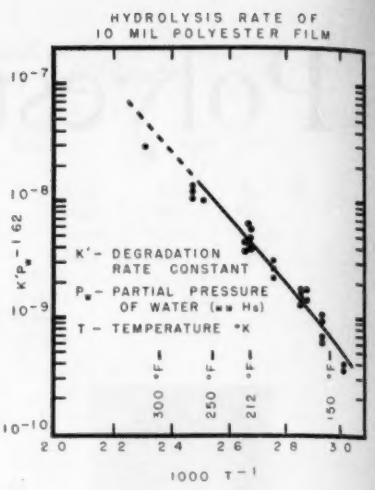


Fig. 2 Hydrolysis rate in straight line relationship to partial pressure of water and temperature

ity, 2 - 3 % shrinkage at 300 F, of this polyester film show that there will be no problem with current leakage or electrical failures due to dimensional changes of the insulation. Under conditions of 90 % relative humidity, although exaggerated for our particular needs, polyester film has an insulation resistance of four orders of magnitude higher than that of rag paper.

The third, perhaps the most important point, is the effect of the system on the polyester film. The component of an hermetic system that has the greatest effect on the useful life of polyester film is moisture. The combined effect of all other components is usually quite small when compared to the effect of water. This is shown by the fact that the time of failure of the film in sealed tube tests can usually be predicted by consideration of hydrolytic degradation alone.

Resistance to hydrocarbons is one of the outstanding characteristics of polyester film. Tests have indicated that in the absence of oxidation products, pure hydrocarbons have conspicuously little effect on this film. Current sealed tube tests are confirming these earlier tests since there has been no embrittlement of the film after exposure to a naphthenic refrigeration oil at 300 F for 180 days.

Polyester film also appears to be resistant to Refrigerant-12 and Refrigerant-22 at the maximum temperatures recommended for these refrigerants in hermetic systems. Thus, this film did not become embrittled during six months exposure to dry Refrigerant-12 at 250 F nor did it become embrittled after four months exposure to dry Refrigerant-22 at 300 F in sealed tube tests.

Returning to the effect of water on polyester film, an extensive study of its hydrolysis rate at various temperatures and humidities shows that the partial pressure of water has a marked effect on its degradation rate. This is shown in Fig. 1 and is in general agreement with experience on other polymers subjected to hydrolytic degradation. Fig. 1 compares the partial pressure of water with the degradation rate constant of 10 mil polyester film at temperatures indicated and humidities of 20% to 100% relative humidity. The average slope of the curves in Fig. 1 is 1.62.

Since it was determined empirically that the hydrolysis rate of this 10 mil film is proportional to the partial pressure of water (P_w) raised to the 1.62 power, it was then possible to obtain a straight line relationship between hydrolysis rate, partial pressure of water, and temperature. This is shown in Fig. 2.

The validity of the relationship shown in Fig. 2 is enhanced when it is realized that the data shown have been obtained from several sources. In all cases, ranging from 150 to 310 F, additional data were found to be consistent with data previously received.

With the use of the degradation rate constant in Fig. 2 it is possible to determine the time that polyester film can be used at various temperatures and water partial pressures before the "initial brittle point" is reached. The results of these calculations are shown graphically for several temperatures of interest in the hermetic motor field (Fig. 3).

The "initial brittle point" is the point at which the film has a 45% reduction in molecular weight. Clarifying further, it is the point at which the polyester film has a loss of tensile strength from approximately 21,000 to 14,000 psi and will fracture when sharply folded. Also, there is a loss of 5 to 10% in electrical properties thereby reducing its dielectric strength to approximately 20,000 volt.

From Dalton's Law of partial pressures, we may determine the partial pressure of water at various conditions (Fig. 4). This is based on the assumption of perfect gas behavior at the low water concentrations indicated. It is also assumed that concentration of water to refrigerant is equivalent in both liquid and vapor phases, since this is closely approached in a closed hermetic system under operating conditions.*

From Figs. 3 and 4 it is possible to determine the time for polyester film to reach the initial brittle point. For example, with the usual maximum water content of production units (100 ppm in refrigerant), this film would be expected to last in excess of five years continuous

*W. R. Briskin, "Moisture Migration in Hermetic Refrigeration Systems", *Refrigerating Engineering*, July 1955, pgs. 42-49.

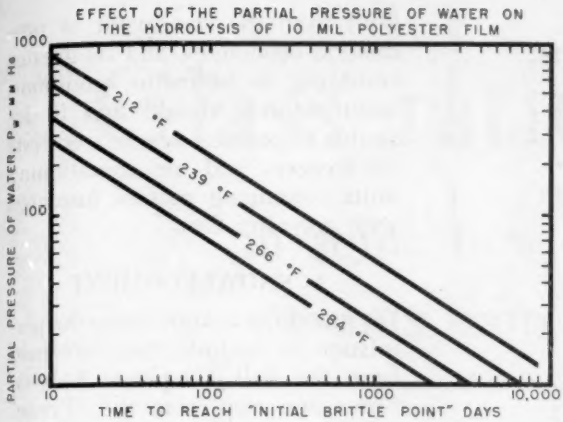


Fig. 3 Effect of the partial pressure of water on hydrolysis

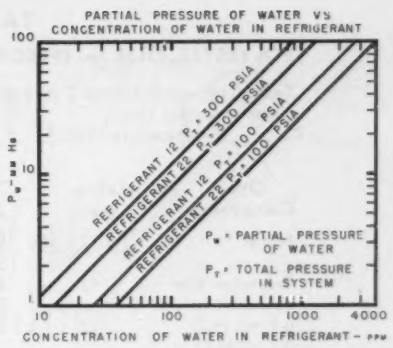

PARTIAL PRESSURE OF WATER VS CONCENTRATION OF WATER IN REFRIGERANT
100
10
1
P_w - mm Hg
REFRIGERANT 12 P_T = 300 PSIA
REFRIGERANT 22 P_T = 300 PSIA
REFRIGERANT 12 P_T = 100 PSIA
REFRIGERANT 22 P_T = 100 PSIA
P_w = PARTIAL PRESSURE OF WATER
P_T = TOTAL PRESSURE IN SYSTEM
10
100
1000
4000
CONCENTRATION OF WATER IN REFRIGERANT - ppm

Fig. 4 Partial pressure of water vs. the concentration of water in the refrigerant

operation at 285 F and 300 psia. A second example would be an exaggerated condition where Refrigerant-22 became saturated with water (1,000 ppm). Under these conditions, the film would be expected to last longer than five years in the normal operation of an air conditioner. Life in a refrigerator or freezer unit should be much longer because of the low solubility of water in the refrigerant at the temperature of the evaporator coil.

ACCELERATED TESTS APPLIED

Although actual operation of a unit is the best test, the expense and time required for such tests has led to the use of several accelerated tests. It is generally recognized that none of these accelerated tests can exactly duplicate the operating conditions in an hermetic system. However, it should be possible to obtain a close estimate of the useful life of an insulating material through the interpolation and extrapolation of data obtained in specially designed experiments.

Figs. 1* through 3 show the importance of the partial pressure of water on the degradation rate of polyester film. Using these data, we examined the conditions used in sealed tube tests containing polyester film, water, refrigerant and other components which were run in several laboratories and found that in every case the partial pressure of water was much greater than that found in operating hermetic systems. This was due to the large amount of added water, the presence of water in the film, and the high refrigerant pressure.

Since pressure has a marked effect on degradation rates, it is important to design experiments which will have pressures similar to those found in actual operating conditions. It is difficult to obtain exact pressures in sealed tube tests. However, pressures can be closely approximated by using the specific volume of the refrigerant at the operating conditions found in the specific hermetic system.

The total amount of water in a sealed tube must be known in order to determine the degradation rate constant for the film. Water is present in the film, in the refrigeration oil, and the refrigerant as well as being on the various surface to 220 F in a vacuum or mosphere. Most of the surface water is removed by heating the surface to 220 F. in a vacuum or in dry air for one hour. Refrigerant normally is supplied with less than 10 ppm water; this nominal value should be included in any calculations. Refrigeration oil may be supplied with as much as 50 ppm water, although it is frequently dried to as low as 20 ppm water for test purposes. Most sealed tube tests have been conducted with the values mentioned above.

However, in the past, polyester film has usually been inadequately dried and the water remaining in the film has resulted in a high partial pressure of water in sealed tube tests due to their design. Thus, the ratio of the weight of film to the volume of the sealed tube has usually been much greater than the ratio of 0.5 lb/cu ft (or 8 gram/1,000 cc) which is representative of many hermetic systems. This means that the effect of the moisture in the film is much greater than it would be in a commercial system.

Polyester film contains up to 0.8% water when stored at usual room conditions. The removal of all this water is quite difficult, e.g. 10 mil film still contains 0.2% water after drying at 20 micron pressure at 100 C. for 2 hr. Until sufficient desorption rate data are obtained, it is recommended that the film be dried at least as vigorously as hermetic motors are for production. Even then there may be enough water present to accelerate a sealed tube test as pointed out in the previous paragraph. However, it is felt that production drying techniques should reduce the water content to less that 0.05%; and that amount should have little effect on a properly designed sealed tube test (see Table II-C). The use of Figs. 3 and 4 shows that in test cylinders of equal volume, tube II-A would result in the embrittlement of polyester film 133 times faster than tube II-C at the same temperature.

When it is desired to embrittle polyester film in a sealed tube test, it is necessary to have at least one milligram of water present for each gram of film. A smaller amount of water would be insufficient to react with the number of ester links necessary to reduce the number average molecular weight by 45% and thus cause embrittlement.

Testing of polyester film in a circulating air oven can give misleading results when evaluating the film for hermetic systems. The major factor is the ready availability of oxygen which appears to have a synergistic effect on hydrolysis of the film resulting in a higher

*Based on data shown in the article "Degradation Studies of Polyethylene Terephthalate" by W. McMahon, et. al., *Journal of Chemical and Engineering Data*, January 1959, pgs. 57-79.

TABLE I

POLYESTER FILM — EFFECT ON HERMETIC COMPONENTS

Sealed tubes contained 2 ml refrigerant, 2 ml naphthenic oil, copper strip, and steel strip.
Temperature: 250 F. Duration: 269 Days

Other Components	Refrigerant	Days To* 1	2	3	4	Penetration (in/mo × 10⁶) Copper	Steel	mg** Chloride Ion
None	12	20	52	267	—	3	7	5.8
		—	150	267	—	3	5	5.3
Polyester film	12	—	40	224	—	3	8	5.4
		—	20	—	269	3	4	4.6
0.1 ml H_2O	12	3	4	9	116	0	129	85.0
0.1 ml H_2O + polyester film	12	—	1	4	118	1	61	50.0
None	22	267	—	—	—	1	2	0.015
Polyester film	22	269	—	—	—	0	3	0.012

* Key: 1 = Slight discoloration or tarnish
2 = Slight copper plating or corrosion
3 = Moderate copper plating or corrosion
4 = Heavy copper plating or corrosion

** The chloride ion content increases as the refrigerant is degraded.

degradation rate. Other polymers also exhibit a large change (5X to 50X) in the hydrolysis rate when oxygen is present. Another factor is that the partial pressure of water varies considerably unless the circulating air oven is either located in a constant temperature and humidity room or has an adequate humidity control of the air inlet. If neither of these two conditions prevails, a variation of 22X can be obtained in the number of days before polyester film becomes brittle (based on the extremes of summer and winter conditions: 80% R.H. at 90 F and 20% R.H. at 73 F) when using this type of test.

Operation of hermetic systems under accelerated load conditions is, of course, the best test to run once the other tests have shown the material has a good possibility of success. Since this new material appears to be so promising, both from the standpoint of production economies, and its greater resistance to hermetic conditions, manufacturers should find it desirable to perform accelerated tests on freezers and air conditioning units containing motors insulated with polyester film.

ACKNOWLEDGMENT

We gratefully acknowledge the permission to include data obtained from the Bell Telephone Laboratories and also from the "Freon" Products Laboratory of the duPont Company. In addition we would like to thank the many people in the refrigeration industry who have given us the advice and information which were needed to make this special study of the applicability of polyester film for hermetic motors.

TABLE II

MILLIGRAMS OF WATER ADDED TO SYSTEM BY VARIOUS COMPONENTS

Component	Method of Calculation	Typical Values (mg.) A	B	C
Polyester film	Gms of film × % water in film × 10	4.00	.50	.10
Refrigeration oil	Gms. of oil × ppm water in oil × 10^{-3}	.20	.10	.10
Refrigerant	Gms. of refrigerant × ppm water in refrigerant × 10^{-3}	.02	.02	.02
Clean metallic surface	Area of Surface (in^2) × 4 × 10^{-4}	.006	—	—
Rusty steel surface	Area of Surface (in^2) × 8 × 10^{-3}	—	.12	—
Total (mg)		4.226	.74	0.22
(ppm Water in Refrigerant)		2113	370	110

Conditions:

A—1.0 gm polyester film (0.4% water), 4 gm of oil (50 ppm water), 2 gm of refrigerant (10 ppm water), 15 in^2 clean metallic surface.

B—0.5 gm polyester film (0.1% water), 2 gm of oil (50 ppm water), 2 gm of refrigerant (10 ppm water), 15 in^2 rusty steel surface.

C—0.2 gm polyester film (0.05% water), 2 gm of oil (50 ppm water), 2 gm of refrigerant (10 ppm water), 15 in^2 clean metallic surface (dried).

WORKERS WARM, BUILDING COLD

Gas-fired, infra-red heat generators installed at the plant of the Junkin Safety Appliance Company, Louisville, Ky., are suspended along one side of a 50 x 100-ft building to radiate over a 25 x 85-ft area. Replacing two forced air space heaters with an input of 185,000 Btu/hr but without adequate output, this equipment consists of seven heaters rated at 168,000 Btu/hr. These Perfection Industries units maintain a thermostat temperature of 70 F in the concrete block, wood-roofed building with outside temperatures within a few degrees of freezing.

Under colder outside conditions, the generators operate continuously and inside levels may drop to 50 F without impaired worker comfort since the radiant energy, developed at 1600 F, is readily and quickly absorbed by both materials and people.

2200 SQ FT OPEN-AIR PATIO IS RADIANT-HEATED ALL YEAR

Copper-tube radiant heating with water at the Marlton Shops in the Crenshaw Center, Los Angeles, Calif., makes it possible to use an outdoor patio for year-round dining and snacking. Tubing is laid on 9-in centers to fit the contour of the 2200 sq ft area. Concrete slab was poured over the tubing. This is a two-zone system which permits shut-off of one side of the terrace when it is not needed.

Glass ratios near 75%

in some new buildings and that means more air conditioning

There have been many large and attractive commercial office buildings erected during the past decade. Some of more recent design have large expanses of glass in which either the clear or the heat absorbing type have been employed. Glass ratios in a few of the new modern buildings are in excess of 75% (glass equals 75% of the wall area).

It is the purpose of this study to show the effect of glass areas on the air conditioning room sensible cooling load and as a consequence the change in first cost, operating cost and overall owning and operating costs, of air conditioning.

The air conditioning sensible cooling load in a typical peripheral commercial office consists of four major components, i.e.:

1. Solar radiation
2. Transmission
3. Lights and other electrical equipment
4. People

Solar radiation and transmission are external sources of heat. Their effect on the cooling load varies approximately in direct ratio to the glass area. Solar or direct radiation is the largest single heat gain to the office area, consequently many methods for reducing the sun's radiant heat have been utilized. Specifically, various types of external shading devices, heat absorbing glass sections, internal shading devices and combinations of these have been used to minimize this excessive heat gain.

Figs. 1 and 2 show a peripheral area floor plan in a commercial office building with a bay dimension of 20 x 17 ft by 9 ft 6 in. floor height. A typical peripheral office in this bay may be 10 x 13 ft. This is the module size used in this study for comparison of the relative air-conditioning loads which result from changing glass areas. In both modules the glass section is assumed to be ¼-in. thickness clear plate with inside venetian blinds, the internal lighting load at 3 watt per sq ft and the people density at 100 sq ft per person.

This report was prepared by the Joint Cooperative Committee of ASHAE and the American Institute of Architects. It has been presented similarly in the Journal of the AIA.

Fig. 1 Office space with 24.5 glass ratio (per cent)

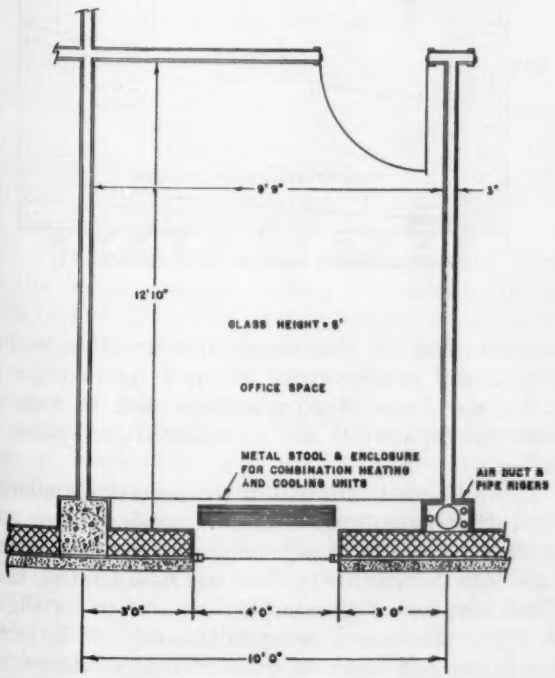
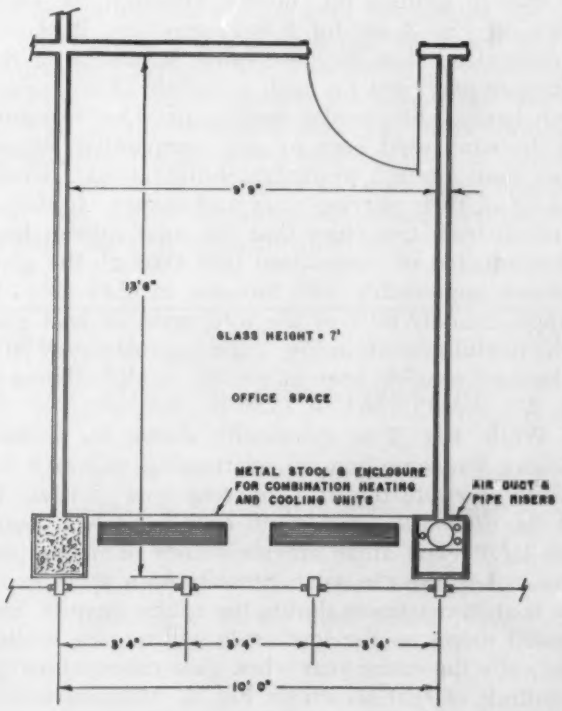


Fig. 2 Office space with 74.0 glass ratio (per cent)



The glass height in Fig. 1 is 5 ft. The stool height for this module is 2 ft 6 in. and the glass ratio is 24.5%.

The glass height in Fig. 2 is 7 ft and the stool height is 1 ft. The glass ratio in this module is 74%.

Fig. 3 shows a breakdown of the air conditioning sensible cooling load at design and also a summation of the components of this load. These values are for the modules shown in Figs. 1 and 2 when oriented

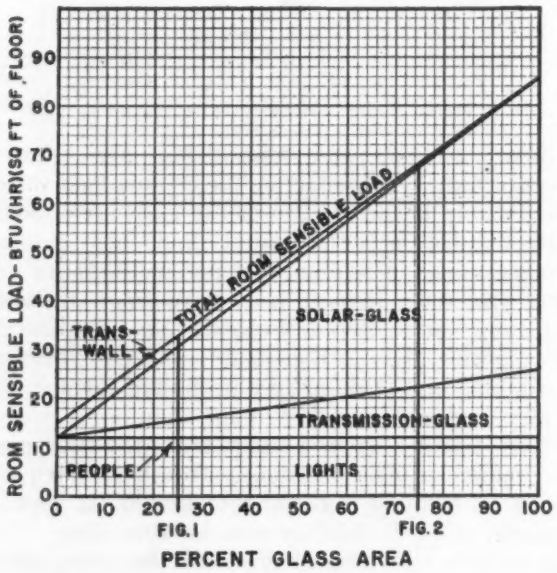


Fig. 3 Room load as related to per cent glass area. This is for west exposure, 40° N latitude, 4:00 p.m., 95 F DB out and 75 F DB in. Room dimensions, 15 ft deep x 9 ft 9 in. wide x 8 ft 10 in. high. White venetian blind on window

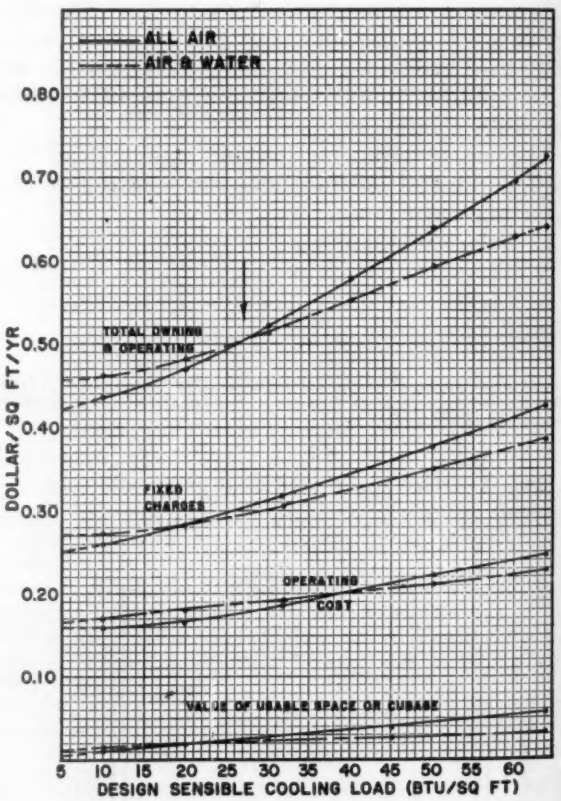
for a west exposure and at 4:00 p.m. of a design summer day in latitude 40° north. Although the loads shown on Fig. 3 are for a west exposure, it should be recognized that they are fairly indicative of the cooling requirement for such a module of any orientation having direct solar heat gains. One exposure only is being used here to give comparative figures rather than attempt to analyze buildings as a whole because of their varying sizes and shapes. It should be noted from this chart that the solar radiant heat and conducted or transmitted heat through the glass increases appreciably with increase in glass area. It is approximately 60% of the total sensible heat gain to the module shown in Fig. 1 and approximately 80% of the total sensible heat gain to the module shown in Fig. 2.

While Fig. 3 is specifically shown for a west exposure the same general relationship will exist for a south exposure but at a different time of year. In fact the difficulties on a south exposure which result from large glass areas are sometimes of major proportions because the solar intensity on a south exposure is at its maximum during the winter months. Sun exposed rooms on the south side will require cooling practically the entire year when glass ratios are in the magnitude of that shown on Fig. 2. Modules with a south exposure and shaded, as for example by adjacent buildings, will require heating during the winter season. Such a widely varying load on one exposure cannot be handled satisfactorily with the conventional "zoning by exposure."

The percent glass ratio for the two modules is the abscissa in Fig. 3. Intersection of the vertical glass ratio lines with the sloping lines showing the major components of the load will indicate the magnitude of each for the various office modules. These data have been tabulated in Table I.

An analysis of the data shown in Table I will indicate a ratio of approximately 300% between Figs. 1 and 2 for the solar radiation component of the sensible load – i.e., 15.1 to 45.8 (column No. 3). For the office modules under consideration, the difference in overall design room sensible load of offices similar to Figs. 1 and 2 is approximately 200% – that is, 33 to

Fig. 4 Total costs, air-water and all-air systems as related to design room sensible load per sq ft



68 (column No. 7). This means that an office similar to Fig. 2 will require twice as much as an office in Fig. 1 if an "all-air" type system is used, or units of double the capacity if an "air-water" type system is installed.

Table II is a tabulation of the percentage change in first cost, operating cost and total owning and operating costs with change in glass areas. Assuming the module shown in Fig. 1 as the base (100%), then the first cost for an air conditioning system for Fig. 2 with 74% glass area is approximately 33% more. These ratios are shown in column 3. Operating costs

increase appreciably as a result of the greater cooling requirements as shown in column 4 of Table II. The significant figure is the percentage increase in owning and operating cost as shown in column 5. Again, assuming module as shown in Fig. 1 as the base (100%), the increase in owning and operating costs of air conditioning for an office as shown in Fig. 2 is approximately 30%.

Table II also shows the approximate costs in dollars for installation of air conditioning (column 6), the resultant fixed charges (column 7), the operating cost (column 8), and the total owning and operating cost (column 9). These are average costs for metropolitan areas in the north. They are also the average cost for certain types of air conditioning systems most often used in the large multi-story, multi-room type building.

This study shows only the variation in the sensible cooling load which results from changes in glass area and for which there is a definite correlation. This is not the case for heating, consequently the heating cost variation has been excluded. Specifically, the solar radiation effect, which is the major variable, is substantially the same for latitudes 30° north to 40° north or for the entire United States. Heating costs are more directly related to areas of the country or latitudes, hence for buildings of similar type and size, the cost variation between southern and northern latitudes is appreciable. It follows, therefore, that heating costs are affected to a considerable degree by the winter outside design temperature as well as glass area. Cost of heating for commercial office

TABLE I

TYPE	GLASS RATIO (%)	DESIGN COOLING LOAD BTU/HR/SQ FT (FLR)					DESIGN COOLING CAPACITY (ROOM)(%)
		SOLAR	TRANS.	PEO.	LIGHTS	TOTAL	
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
FIG. 1	24.5	15.1	6.0	1.7	10.2	33	100
FIG. 2	74.0	45.8	10.3	1.7	10.2	68	205

buildings of similar design may vary from a low of $0.03/ft²/yr for the south to $0.13/ft²/yr for those in the northern section of the United States.

Fig. 4 shows the variation in costs with changes in the design sensible cooling load (change in glass ratio) for an "air-water" system and an "all-air" system.

An "air-water" type system basically is one in which these two media are used for the extraction from or the addition to the conditioned space of both sensible and latent heat. An "all-air" type system is one in which only air is used to perform the above functions. More specifically, an "air-water" type system may be an induction unit system with the induction units located under the windows as shown in Figs. 1 and 2 or a fan unit system with supplementary primary air and the units similarly located. An "all-air" type system may be the double duct type with air blender located under the window and the two air supply ducts in riser chases somewhat as shown.

This chart shows the relative costs of these two basic systems. For example, the "all-air" type system will require more useable space or cubage for the air treating assembly and for distribution to the conditioned areas. The operating costs will be higher at higher loadings (larger glass areas) because of the large volumes of air required to absorb the sensible heat, and the fixed charges will be higher because of the greater initial investment.

Total owning and operating cost data from Table II is an average for various types of systems while data shown in Fig. 4 is for two specific types of air conditioning. The fixed charges, operating cost and

TABLE II

TYPE	GLASS RATIO (%)	FIRST COST RATIO (%)	OPERATING COST RATIO (%)	TOTAL OWNING AND OPERATING COST RATIO (%)	FIRST COST $	FIXED CHARGES $	OPERATING COST $	TOTAL OWNING AND OPERATING COST $
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
FIG. 1	24.5	100	100	100	4.10	0.315	0.190	0.505
FIG. 2	74.0	133	126	130	5.45	0.420	0.240	0.660

overall cost figures, therefore, are not exactly comparable. An office module with a design sensible cooling load of 53 Btu/ft²/hr will have a total owning cost of $.60 per sq ft per yr with an "air-water" type system and a total owning cost of $.65 per sq ft per yr if an "all-air" type system is installed.

The inside area or the interior zone, as it is often called, is not affected by the glass areas of the exterior or peripheral zone. The year-round cooling of this area is a completely separate problem and not considered in this study. The heat gain to the interior area consists of lights, electrical equipment and people and is a stable cooling load throughout the year.

In summary, therefore, it may be stated that the larger the glass area, the greater the air conditioning load and, as a consequence, the higher the first cost as well as the operating cost for air conditioning. Specifically, for these two type modules as shown in Figs. 1 and 2, the difference in owning and operating costs for the air conditioning system, exclusive of heating costs, is approximately 30%.

ASHRAE TO MEET AT LAKE PLACID

First of the annual meetings of the American Society of Heating, Refrigerating and Air Conditioning Engineers will be held at the Lake Placid Club, Lake Placid, N. Y., June 22-24, 1959.

Standards—

an all-industry problem

A. T. BOGGS III
ASHRAE Technical Secretary

There are a number of available vehicles for the development and promulgation of required standards. Here is a summary of four areas in the following classifications:

1. ASHRAE standards in effect and proposed
2. Intersociety standards
3. Cooperation with government agencies
4. International aspect of standards

ASHRAE STANDARDS

Prior to the merger of ASRE and ASHAE, the development of standards by the single societies was a relatively simple procedure in that ASRE was primarily interested in those standards related to that part of the temperature scale covering air conditioning down through the lowest temperature field of cryogenics. ASHAE, on the other hand, was primarily interested in that segment of the temperature scale covering air conditioning through the high temperature applications of heating. At the present time, the total of existing standards, those nearing completion, and ASHRAE-proposed numbers 46. It is obvious that a new system of reference to individual standards must be developed. This is one of the immediate problems to be discussed by your Standards Committee.

Our Society standards may be divided as: heating, refrigerating, air conditioning, ventilating and general. For example, using this method of segregation, the Society now has 5 standards in effect or being developed in the heating area, 21 in the refrigerating area, 8 related to air conditioning, 5 allied with ventilation problems, and 7 covering general subjects. It is clear that ASHRAE will play an important part in the standardization program of American industry. A complete listing of the effective standards and those being developed along with the committee personnel responsible will appear in a future issue of your JOURNAL.

INTERSOCIETY STANDARDS

Although the previous paragraph indicates the degree of standardization activities by the Society, of equal importance are the efforts of Society representatives on the many standardization committees of allied industry groups.

1 – The best-known standardization group in the national and international standardization fields is the **American Standards Association (ASA).** ASHRAE is represented on 29 sectional committees of ASA. To indicate the degree of common interest in the standardization area, there are 11 of these sectional committees that had representatives from ASRE and ASHAE. Eight of the sectional committees are sponsored by ASHRAE. A-114, Thermal Insulating Materials, and Z-74 on Performance of Effluent and Gas Cleaning Equipment, were sponsored by ASHAE and will continue under the sponsorship of the new Society. Six sectional committees, B9 Safety Code for Refrigeration, B38 Household Refrigerators, B53 Refrigeration Terms & Definitions, B59 Marine Refrigeration Installations, B60 Refrigerant Expansion Valves, and B79 Designation of Refrigerants were sponsored by ASRE and will continue under ASHRAE.

2 – There are several standards committees of the **American Society for Testing Materials (ASTM)** that have representatives from ASHRAE. At the present time 8 ASTM committees include Society representatives. One of these, C16 – concerning Thermal Insulating Materials – previously had representatives from both ASHAE and ASRE. Three ASTM committees concerning Acoustical Materials, Gaseous Fuels, and Atmospheric Sampling and Analysis included ASHAE representations while four other committees related to Corrosion of Materials, Petroleum Products and Lubricants, and Industrial Water, had representatives from ASRE. These committees will continue to have ASHRAE representation.

3 – The **National Fire Protection Association (NFPA)** had nine standardization groups which included ASHAE representatives. These committees will continue with ASHRAE representation and upon review, it is likely that additional NFPA standardization groups will include Society representation. The aspect of safety in all industry standards will be an area of principal interest for your Standards Committee and liaison activities on intersociety safety standards committees such as NFPA will be continued.

4 – Two Power Test Code committees of the **American Society of Mechanical Engineers (ASME)** relative to Safety and Relief Valves

(*Continued on page* 112)

"There is reasonable evidence to assume that the American Economy could save $4 billion yearly through more and better standards."—H. T. Hallowell, Jr., president, American Standards Association, in keynote address at the Ninth National Conference on Standards.

Heat transfer rates

between heated tubes and a boiling refrigerant

C. F. STALEY

MERL BAKER
Member ASHRAE

The object of this research was to obtain experimental data for Refrigerant-22 and to correlate the data in terms of the physical properties of the refrigerant and fundamental parameters of the evaporator systems.

At the present time there is no exact mathematical method for the determination of the inside conductance for boiling refrigerants flowing within a horizontal tube. This problem is both difficult and complex due to the lack of knowledge of the mechanism of boiling heat transfer. Studies in this important field generally fall into two broad categories; one is experimental determinations of coefficients for use as practical design data, and the second is the formulation of a general correlation based on fundamental parameters.

The superimposing of pool boiling upon forced convection heat transfer makes correlation of experimental data difficult because it is influenced by many factors. In addition to the properties that must be considered for forced convection the surface tension, latent heat of vaporization, saturation temperature, vapor density, mass flow, geometry of flow and the characteristics of the surface transferring the heat must be considered. This requires, for the present at least, a separate investigation for each combination of fluid, surface and geometry.[1]

Charles F. Staley, formerly a research assistant at the University of Kentucky, is a mechanical engineer with Pratt & Whitney. Merl Baker is Executive Director, Kentucky Research Foundation and Professor of Mechanical Engineering, University of Kentucky.
This paper was presented as "Heat Transfer Rates from Heated Horizontal Tubes to Boiling Refrigerant—22" at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958.

BACKGROUND

It is well known that data may be correlated for non-boiling or single-phase forced convection heat transfer b ythe use of an equation of the form,

$$(1) \qquad Nu = C\,Re^{m}Pr^{n}$$

Where, Nu = Nusselt number
Re = Reynolds number
Pr = Prandtl number
C = Constant

In a simultaneously boiling and flowing liquid, the heat transfer rate may be much greater than during single phase forced convection.

Rohsenow and Clark,[12] in a paper published in 1951, developed a correlating equation for pool boiling. With the aid of high-speed motion pictures they compared the actual heat transfer required to form bubbles of various sizes on the heating surface with the difference between boiling and single-phase forced convection heat transfer rates. They found however, that the heat transferred to the bubbles accounts for an exceedingly small, in fact practically negligible, part of the difference, and therefore stated that the bubbles do not act as important carriers of heat for the heat transfer process during boiling.

Rohsenow and Clark further observed that a characteristic of the bubble path is a large bubble velocity perpendicular to the tube wall compared with the mean fluid velocity, thus indicating that relatively high velocities are induced in the otherwise dormant fluid adjacent to the heating surface.

It was then propounded that the high heat transfer rate associated with surface boiling is due primarily to the violent agitation caused by bubble motion in the region adjacent to the heating surface, and that the fluid velocity has little effect when compared to bubble velocity.[12] Hence the effect of Reynolds number, based on fluid velocity and pipe diameter, became less and less as boiling increased. Experimental data with water by Rohsenow and Clark bore out these contentions.

In a later paper published in 1952, Rohsenow[13] reasoned that a correlation of boiling heat transfer data should contain a bubble Reynolds number based on bubble diameter and velocity and a bubble Nusselt number based on bubble diameter. Also the correlation should contain the normal Prandtl number, since most of the heat transfer goes directly to the liquid.

By defining the bubble Reynolds number as,

$$(2) \qquad Re, b = \frac{G_b D_b}{\mu_L}$$

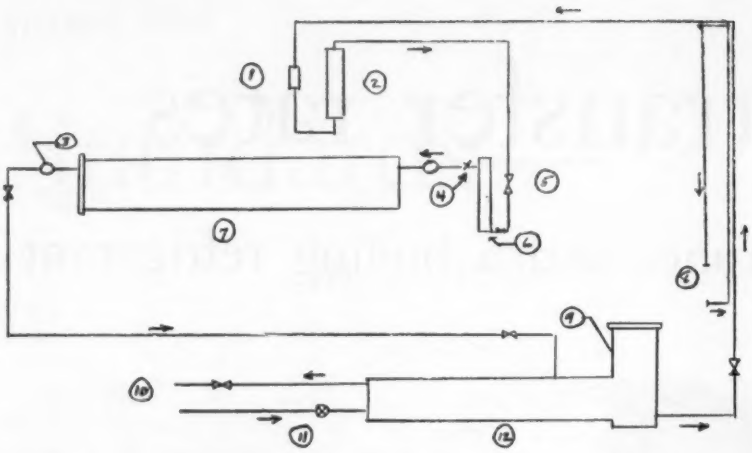


Fig. 1 Schematic flow diagram of test apparatus. (1) strainer, (2) flowrater, (3) sight gage, (4) thermocouple, (5) flow control, (6) preheater, (7) evaporator assembly, (8) suction and discharge to cooling unit, (9) pump, (10) suction and discharge to cooling system, (11) expansion valve, (12) test section

where,

G_b = Mass velocity of bubbles at their departures from surface, lb_m/ft^2hr
D_b = Diameter of bubble as it leaves surface, ft
μ_L = Viscosity of saturated liquid, $lb_m/ft\ hr$

and using a bubble Nusselt number defined as,

$$(3) \qquad Nu, b = \frac{h\,D_b}{k_l} \quad *$$

where,

h = Surface conductance, Btu/hr ft F
k_l = Thermal conductivity Btu/hr ft F

Rohsenow, using equations by others for bubble characteristics, developed the following equation,

$$(4) \qquad \frac{c_L\,\Delta t}{\lambda} = C\left[\frac{q''}{\mu_L\,\lambda}\sqrt{\frac{\sigma}{g\,(\rho_L-\rho_v)}}\right]^n\left[\frac{c_L\,\mu_L}{K_L}\right]^{1.7}$$

where,

c_L = Liquid specific heat Btu/lb_m F
Δt = Temperature difference between tube surface and fluid F
λ = Latent heat of evaporation Btu/lb_m
q'' = Heat flux density Btu/hr ft^2
μ_L = Liquid dynamic viscosity $lb_m/ft\ hr$
σ = Surface tension of refrigerant lb_m/hr^2
g = Gravitational constant ft/hr^2
ρ_L = Liquid density of refrigerant ft^3/lb_m
ρ_v = Vapor density of refrigerant ft^3/lb_m
k_L = Liquid thermal conductivity Btu/hr ft F
C = A constant depending on surface
n = Exponent to be determined.

which he used to correlate data from a number of investigators.

The above equation is dimensionless when consistent units are used. Rohsenow believed that the equation could be used for both convection, where pipe Reynolds number is unimportant, and for pool boiling.

The constant C will necessarily depend on the fluid and heating surface combination, as values for the surface tensions for liquid-surface, vapor-surface were omitted (for lack of information in the determination of the bubble contact angle.)

Equation (4) may be put into the form,

$$Nu, b = \frac{1}{C}\,(Re, b)^{1-n}\,(Pr)^{-0.7}$$

for comparison with equation (1).

The exponent "n" may be determined from a log-log plot of

$$\frac{C_L\,\Delta t}{\lambda}\left[\frac{C_L\,\mu_L}{k_L}\right]^{-1.7} \quad \text{vs.} \quad \frac{q''}{\mu_L\,\lambda}\sqrt{\frac{\sigma}{g\,(\rho_L-\rho_v)}}$$

Several investigators have successfully used this analysis to correlate data for various refrigerants flowing in horizontal tubes. Methyl chloride and Refrigerant-12 were correlated by Baker, Touloukian, and Hawkins[14] with an equation similar to Equation (4) but with the addition of a quality ratio term. In later research with Refrigerant-12, Baker[15] found this quality ratio term was negligible. Data for Refrigerant-11 are successfully correlated by Miller[16] and Baker,[17] and in addition Baker and Miller found that the heat transfer rate may be increased during low flow rates by removing the vapor, during progressive evaporation.

Due to several refrigerants being correlated by Equation (4) or similar equations which seem to bear out the Rohsenow analysis, it therefore seems desirable to extend the limits of the variables.

This present research endeavor proposes to correlate experimental data by Equation (4) for Refrigerant-22 in a temperature range of 32 F to 60 F and the flow rate varying from 60 to 180 lb/hr. No correlation or experimentally determined conductance values for Refrigerant-22 have been found in the literature.

EXPERIMENTAL APPARATUS

The apparatus used in this experiment is essentially a combination of equipment used previously at the University of Kentucky.

The test evaporator is a horizonal 0.625 in. OD, 0.545 in. ID copper tube, 39⅝ in. long. Heat was provided by a 2000-watt immersion type heater. The test tube evaporator passes through a 4-in. pipe containing this primary heater and a charge of Refrigerant-11. The Refrigerant-11 acts as a transfer medium by boiling on the heater and condensing on the test evaporator tube surface. A uniform heat flux is provided to the tube surface since the liquid Refrigerant-11 covers the heater but does not touch the test tube surface. The 4-in. pipe is enclosed within an 8-in. pipe containing a 500-watt immersion heater and another charge of Refrigerant-11. Sufficient energy was supplied to this heater so as to maintain the same temperature in the 8-in. pipe that existed in the 4-in. one, thus

* This was defined by Jakob, Chapter 29, *Heat Transfer*, John Wiley and Sons, 1949

insuring that the total output of the primary heater would be transferred to the test evaporator. Immediately up stream from the test section an immersion type 2000-watt heater was used to vary the quality during tests.

TEST PROCEDURES

No data were taken until the apparatus had been operating for some time and all variables were in equilibrium. Several minutes were required for the refrigeration machine to cool the test refrigerant to the lowest temperature possible which was slightly less than 32 F. After this initial temperature was reached, the immersion pump was started and the flow control valve adjusted to give the desired flow rate. The authors did not observe any flashing in the flowrater as did Baker and Miller. No doubt the trouble was eliminated with the addition of the subcooler following the pump. The degree of sub-cooling entering the test evaporator, if any, could not be exactly controlled, due to the heat loss in the line from the pump to the test section.

As the test refrigerant temperature was established by the balance of the cooling capacity of the refrigeration machine and the heat gains in the test circuit, a higher test refrigerant temperature was obtained by manual adjustment of the gate valve located in the suction line of the refrigeration machine. Of course this was a trial and error adjustment.

After a steady flow was

TABLE I
EXPERIMENTAL DATA FOR REFRIGERANT-22

SERIES A

Test No.	Boiling Temp. F	t F	q" Btu/Hr Ft²	Flow-Rate Lb/Hr	Subcooling F
1	34.4	7.5	1507	58	0.5
2	40.2	9.9	3014	59	—
3	43.2	9.7	3014	60	—
4	35.6	11.4	3985	59	0.8
5	38.7	11.4	3990	61	—
6	40.7	13.4	5507	60	6.3
7	44.9	13.2	5476	60	6.8
8	50.0	13.2	5491	60	9.3
9	48.8	6.1	1507	60	—
10	51.1	5.9	1507	61	—
11	53.4	5.8	1507	59	—
12	45.6	9.6	3014	61	0.2
13	50.0	10.2	3004	61	3.9
14	53.1	9.7	3021	61	2.2
15	54.7	9.5	3014	61	—
16	45.6	10.8	4000	60	—
17	47.2	11.0	4014	59	—
18	49.0	11.0	4000	60	—
19	50.5	11.3	4010	60	—
20	44.5	12.2	5507	61	1.5
21	52.2	12.1	5507	62	—
22	53.2	12.2	5507	62	0.1
23	56.0	12.1	5507	64	0.3
24	45.4	14.4	7203	62	8.5
25	50.0	13.3	7217	61	8.8

SERIES B

Test No.	Boiling Temp. F	t F	q" Btu/Hr Ft²	Flow-Rate Lb/Hr	Subcooling F
1	35.2	6.9	1493	92	0.4
2	36.8	6.9	1504	88	0.3
3	36.8	9.6	3113	91	—
4	40.1	9.8	3005	90	0.1
5	36.0	10.3	3984	91	2.6
6	40.6	10.6	4000	90	3.5
7	35.8	12.6	5507	90	4.6
8	39.6	13.1	5478	90	7.2
9	43.5	12.2	5507	93	9.1
10	46.6	12.4	5507	90	11.2
11	50.8	12.1	5493	91	13.8
12	49.9	5.8	1507	92	—
13	52.4	5.5	1507	89	—
14	47.8	9.4	3007	90	—
15	49.5	9.1	3014	91	—
16	49.5	9.1	3014	91	—
17	51.6	8.8	3014	90	—
18	53.4	8.8	3014	90	—
19	47.4	10.4	4000	91	1.5
20	52.1	9.7	3995	91	—
21	54.6	9.6	4000	90	—
22	49.8	12.3	5510	91	—
23	55.8	12.0	5522	90	—
24	44.9	14.1	7246	89	13.5
25	51.9	13.4	7246	90	15.1

SERIES C

Test No.	Boiling Temp. F	t F	q" Btu/Hr Ft²	Flow-Rate Lb/Hr	Subcooling F
1	34.9	6.1	1507	121	—
2	36.1	6.1	1507	120	0.2
3	33.1	8.9	3014	122	—
4	35.2	8.9	3014	122	—
5	38.1	9.5	4000	123	—
6	43.2	9.7	3986	122	4.0
7	42.4	10.6	5478	120	8.7
8	47.0	10.7	5492	119	10.2
9	50.6	11.2	5507	120	11.8
10	47.4	5.9	1507	119	—
11	48.8	5.5	1507	121	—
12	49.9	5.4	1507	121	—
13	50.6	7.8	3014	122	—
14	49.6	9.7	4029	121	4.6
15	52.3	9.6	4019	121	5.2
16	47.0	11.1	5478	120	—
17	49.6	11.0	5507	120	0.3
18	54.2	10.8	5493	120	1.1
19	57.4	10.6	5508	120	3.3
20	45.2	12.9	7246	121	3.9
21	49.7	13.1	7261	122	4.6
22	52.6	12.4	7246	123	6.3
23	56.8	12.1	7232	120	8.0
24	56.8	11.8	7246	120	8.9
25	60.1	11.8	7246	120	9.2

SERIES D

Test No.	Boiling Temp. F	t F	q" Btu/Hr Ft²	Flow-Rate Lb/Hr	Subcooling F
1	33.7	5.5	1507	149	—
2	35.3	5.5	1507	149	—
3	33.4	8.0	3014	151	—
4	35.4	7.8	3014	152	—
5	32.0	9.7	4000	152	—
6	34.4	10.4	4000	154	—
7	38.2	10.7	4000	148	—
8	40.3	10.0	4014	151	—
9	42.8	11.6	5507	149	9.6
10	45.3	11.2	5507	152	11.4
11	50.3	11.0	5492	150	13.8
12	49.0	5.1	1507	151	2.2
13	50.8	5.0	1507	152	2.1
14	50.9	7.5	3014	151	—
15	53.6	7.6	3014	150	—
16	51.2	9.0	4000	150	—
17	51.7	8.4	4029	150	—
18	50.4	10.9	5478	152	—
19	54.3	11.0	5507	149	0.6
20	42.2	13.1	7217	150	4.1
21	46.5	12.5	7246	150	5.3
22	50.6	12.4	7261	151	6.7
23	53.8	11.9	7262	150	7.8
24	56.0	11.4	7203	151	8.4
25	57.7	11.2	7275	153	9.2

SERIES E

Test No.	Boiling Temp. F	t F	q" Btu/Hr Ft²	Flow-Rate Lb/Hr	Subcooling F
1	33.1	5.2	1517	180	—
2	38.7	5.2	1511	180	0.9
3	32.5	8.4	3014	181	—
4	33.4	8.2	3014	181	0.6
5	34.1	8.3	3005	181	—
6	35.2	8.3	3000	181	—
7	38.3	9.8	4005	181	—
8	39.9	9.6	3993	180	—
9	38.8	11.4	5498	181	1.9
10	40.7	11.6	5507	178	2.3
11	49.1	4.9	1507	179	3.2
12	50.7	4.5	1507	180	2.18
13	49.7	7.2	3014	180	5.4
14	50.8	7.1	3009	180	5.6
15	46.4	8.7	4000	180	6.7
16	50.0	8.3	4000	179	7.6
17	53.6	8.5	4000	179	9.3
18	46.6	9.8	5507	180	8.9
19	50.1	9.8	5507	180	10.2
20	54.6	9.8	5478	179	11.9
21	46.1	12.11	7261	180	6.8
22	51.6	12.2	7261	182	7.9
23	55.7	11.2	7232	180	9.8
24	57.7	11.3	7232	180	10.4
25	59.9	11.2	7275	179	11.2
26	61.1	10.6	7217	180	11.5

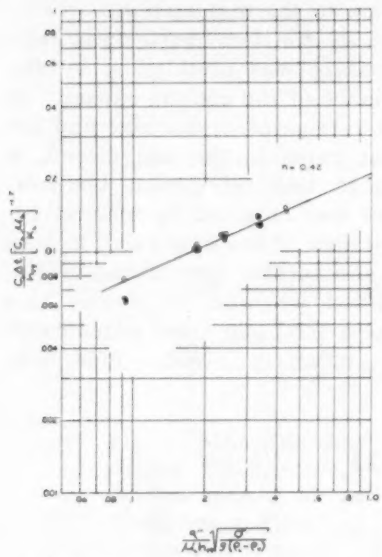


Fig. 2

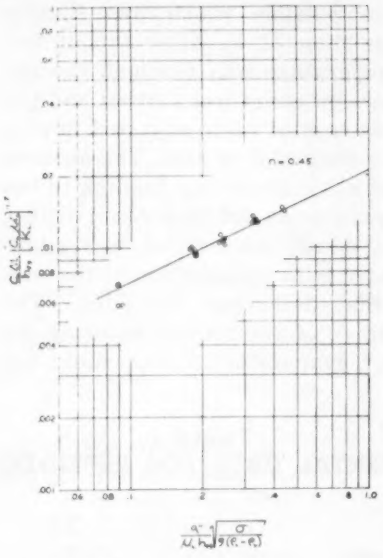


Fig. 3

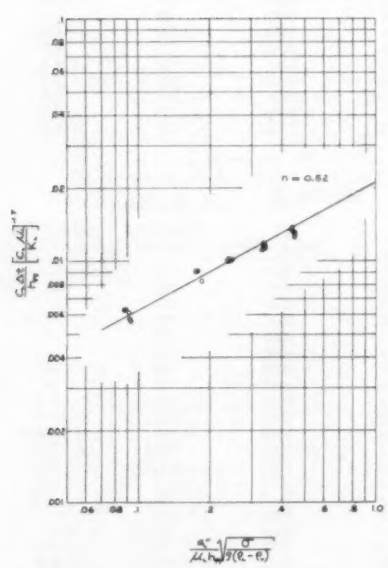


Fig. 4

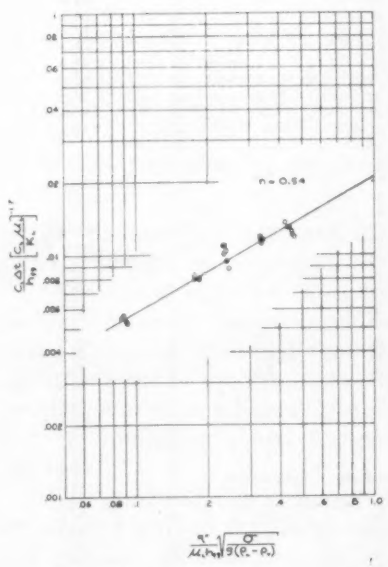


Fig. 5

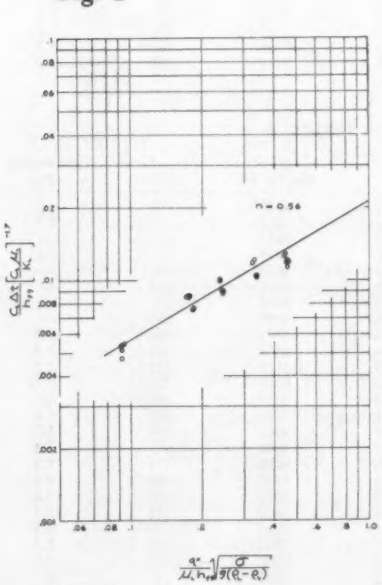


Fig. 6

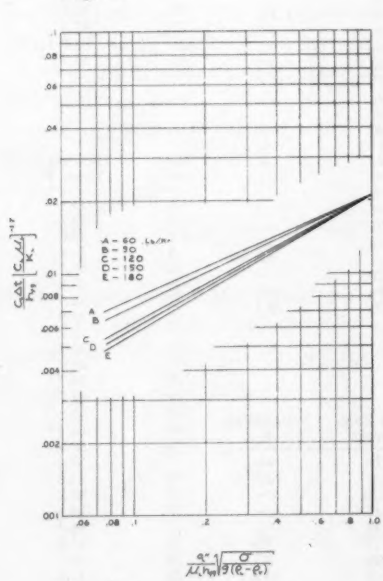


Fig. 7

achieved the primary heater was turned on and the control was adjusted to give a desired heat rate. The guard heater was adjusted as needed throughout the test to maintain temperature equilibrium between the primary and guard Refrigerant-11 baths.

A record was made of the following values in the given order: test refrigerant temperature; temperature entering test evaporator; temperature entering preheater; tube surface temperature; heat input to primary; flow rate; and primary and guard bath temperatures. All temperature readings were recorded in millivolts and the heat input in watts.

A series of tests was selected consisting of points taken with constant primary heater input and flow rate.

Although the tube surface thermocouples were read in parallel to give one record, a slight fluctuation could be observed. This undoubtedly was due to irregular flow of the refrigerant and non-uniform wetting of the evaporator tube.

Data points were taken successively and pertinent values changed but little from one record to the next allowing several points to be averaged, thus minimizing the slight fluctuations. Furthermore a few sample calculations will show that a change of 2 F in the refrigerant temperature does not change the refrigerant properties sufficiently.

The flowrater float indicator oscillated over a small range of $\pm(1)$ lb_m/hr or less, and had a slight tendency to "drift" on some runs from the rate under test. This was due, in all probability, to a small amount of flashing, although not visible, between the flowrater and the test evaporator and to the irregular flow in the evaporator. As a running check was made on the flowrater, the desired rate was maintained by slight adjustments to the needle valve during a test. At no time was flashing of the refrigerant visible in the flowrater when data were being taken.

The data taken are shown in Table-I on the preceding page. Each record shown represents an average of from three to six similar original data points.

CORRELATION AND DISCUSSION

During tests within the low temperature range and the lowest heat flux density (1500 Btu/hr ft²), the primary Refrigerant-11 bath existed at approximately 57 F. Even with the aid of insulation around the 8-in. pipe containing the guard heater, Fig. 2, thermal equilibrium could not be maintained and the guard bath was 6 to 8 F higher than the primary. This additional heat input, to the primary bath surplus to that given by the electric heater, required that appropriate corrections be applied. A rectangular plot of Δt as ordinate versus the temperature difference between the primary and guard, revealed a straight line for all flow rates, thus indicating that the error was directly proportional to the primary and guard temperature difference. Extrapolating to the zero abscissa, a more accurate value of Δt was obtained. Corrected values for Δt are listed in Table I. The correction was applied to test 1 in Series A and tests 1 and 2 in Series B, C, D, and E, only.

Before discussing a correlation of experimental results a few observations can be made from the data given in Table I. The temperature difference between the tube surface and fluid increases with a rise in the heat flux and increases with a decrease in flow rate. These observations are both reasonable and in accord with data of other investigators. It can also be seen that Δt for a given heat flux density and flow rate decreases with an increase in boiling temperature.

The physical properties of the refrigerant were determined at the boiling temperature from references given in the Bibliography. Equation (4) is plotted on log-log coordinates for various flow rates (Figs. 2 through 7).

It can be observed from these plots that each flow rate has a particular slope but all have the same ordinate intercept, which is the constant C in Equation (4). Individual lines representing the respective flow rates most accurately are shown collectively in Fig. 8. For the data presented, the exponent "n" for Equation (4) is different for each flow rate.

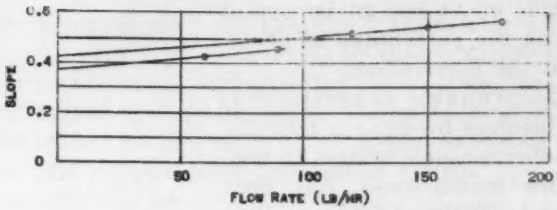


Fig. 8 Dependence of "m" of Fig. 7 with flow rate

The discussion of the previous research in the section on background concluded that liquid in contact with the heating surface, rather than vapor, produced the most effective rate of heat transfer. If this hypothesis is true, the heat flux density should be based on wetted area only, but due to lack of information on the actual surface in contact with the liquid, the heat flux density was necessarily based on the entire inside surface. On the latter basis, for a decrease in flow rate, the wetted area and thus the hydraulic radius is decreased, provided that the liquid is not being entrained. This reasoning was proposed in references 14 and 15 as a possible explanation of a break in the slope "n" in correlating experimental data for Refrigerant-12. Miller[16] found that "n" increased with the flow rate in experiments with Refrigerant-11, therefore some type of variation of "n" with the flow rate might be expected.

Fig. 8 shows a plot of "n", versus the flow rate.

As can be observed from Fig. 8 the slope varies as a straight line with the flow rate except for an apparent break in the range between 90 and 120 lb/hr flow rate. From the graph, the following empirical relationships may be obtained:

(6) $n = 0.00791m + 0.42$ For $m = 120, 150$ or 180

and

(7) $n = 0.00791m + 0.395$ For $m = 60$ or 90

where,

m = Flow rate in lb/hr

The ordinates at zero abscissa in Figs. 3 through 7 are 0.021 and all are equal. The correlating equation takes the form,

$$(8)\qquad \frac{C_L \Delta t}{\lambda} = 0.021 \left[\frac{q''}{\mu_L \lambda \sqrt{\dfrac{\sigma}{g(\rho_L - \rho_v)}}} \right]^n \left[\frac{C_L \mu_L}{k_L} \right]^{1.7}$$

where,

$n = 0.00791m + 0.42 \quad 120 \leq m \leq 180$
$n = 0.00791m + 0.375 \quad 60 \leq m \leq 90$

Table II indicates the accuracy of Equation (8) by showing the per cent deviation of,

$$0.021 \left[\frac{q''}{\mu_L \lambda \sqrt{\dfrac{\sigma}{g(\rho_L - \rho_v)}}} \right]^n$$

from values of

$$\left[\frac{C_L \Delta t}{\lambda} \right] \left[\frac{C_L \mu_L}{k_L} \right]^{1.7}$$

It is noted that the Prandtl number had only a small effect on the correlation. The 1.7 exponent for Prandtl's number suggested by Rohsenow for liquids in contact with clean surfaces, might not be applicable. An attempt was made by the authors to determine if the exponent was different for Refrigerant-22 boiling within the test evaporator. This attempt was inconclusive as it was found for the experimental range that Prandtl's number varied in a narrow range from 1.310 to 1.360 with the values for the majority of the test being from 1.315 to 1.345. Correlating plots were tried for sample flow rates without Prandtl's number term and again little effect was noted except that a slight "damping" of the points at the lower heat rates is achieved by its use. Although it had little effect, the Prandtl number was retained along with the 1.7 exponent for the purpose of generality.

There are several individual effects that possibly might have caused the deviations, but they are rather difficult to separate and identify. Realizing this limitation the possible reason for deviations will be discussed accordingly.

It can be reasoned from Tables I and II and the plots of Fig. 7 that the most critical parameter is the temperature difference between the tube surface and the boiling temperature Δt. It can be observed that a slight change in

the value of Δt has an important effect on the percentage of deviation in the correlations.

The probable experimental error involved by using a potentiometer-thermocouple system in temperature measurement has been discussed. It is the controlling factor in explaining the test deviation since the errors in the measurement of the heat input by precision wattmeter, may be considered negligible.

One of the more obvious factors that could cause deviations is the liquid entering the test evaporator subcooled rather than saturated. Since Equations (4) and (5) have been recommended for an entering liquid at saturated conditions, subcooling may alter these. It might be reasoned that a portion of the tube evaporator is in contact with a subcooled liquid for a definite length while the liquid is being heated to the boiling temperature. It is possible that the length is sufficiently great to reduce the average tube temperature significantly below values resulting from entry of a saturated liquid. Since the refrigerant boiling temperature is read immediately after the test section, subcooling is not reflected in the measured refrigerant temperature. Accordingly, values of Δt in Table I for subcooled points are somewhat lower than actual. The subcooling may also affect the vapor-liquid ratio which could change the characteristics of flow.

In Series E of Table I relatively larger deviations are observed. However it can be noted that the test runs in the higher temperature range for Series E have a higher degree of subcooling than were encountered in other series. The representative line was drawn to take into account both the subcooled points along with the saturated (or quality entering points) thus showing that even with the high degree of subcooling, a satisfactory correlation is achieved.

The effect of dry surfaces on the rate of heat transfer has been discussed and may offer a partial explanation for the exponent of the correlation Equation (8) being dependent on the flow rate. Baker[14,15] accounted for the effect of flow rate in tests with Refrigerant-12 and methyl chloride with an equation similar in form to

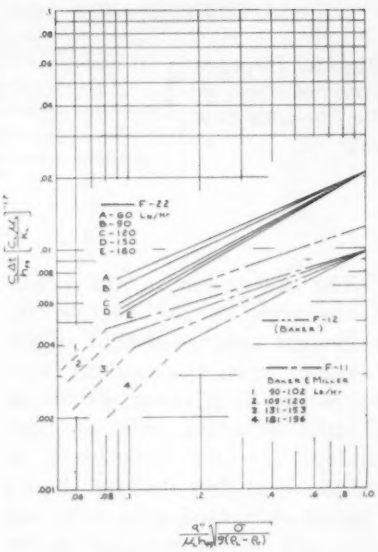


Fig. 9 Comparison of various refrigerants

Equation (4), but with the inclusion of a modified Reynolds number (and a quality ratio term as discussed previously). Baker found that the Reynolds number for forced convection was important for higher flow rates, namely for flow rates of 120 lb/hr and higher. An attempt by the authors to include this pipe Reynolds number for Refrigerant-22 was unsuccessful. Several forms of Reynolds number were used starting from the one suggested in reference 14, but this did not yield satisfactory results. A few sample plots did show that the deviation (of Table II) could be lessened by using pipe Reynolds number in combination with Equation (8) with the variable exponent. This may be an unimportant peculiarity but it could indicate that a further modified Reynolds number taking into account the effect of progressive evaporation might lead to a more general correlation. This would require a separate investigation.

It has been shown that the correlation depends to some extent upon the flow rate, therefore a discussion of the types of flow should logically precede a more extensive analysis of the data.

Simultaneous liquid-vapor flow has been classified as bubbling, stratified, wavy, slugging and annular flow depending on the disturbance of the two phases. Stratified, wavy, irregular wavy, and slugging were observed through the sight gages for various tests in this investigation and undoubtedly bubbling flow occurred within the test evaporator during some runs.

Bubbling flow is characterized by bubbles that can be observed in a liquid phase. This type of flow occurs when the liquid to vapor ratio is high. Stratified flow occurs when the liquid to vapor ratio is decreased, and is characterized by the two phases flowing intact. As the flow rate is increased and the liquid portion is decreased, a condition known as wavy flow occurs in which waves are formed in the liquid-vapor interface. These wave formations can be either of a pattern type or irregular with the latter occurring at increased flow rates until actual entrainment and spraying upward of the liquid by the vapor phase. With further increased flow rates, slugging flow exists. This type is characterized by highly erratic and violent entrainment of the liquid by the vapor and slugs of liquid fill the tube completely but with no pattern.[14] At any one flow rate progressive evaporation could conceivably cause two types of flow to occur within the test evaporator.

Stratified, wavy and slugging flow were observed during the various tests in this experiment. Stratified flow prevailed throughout most of the runs in Series A and B (60 and 90 lb/hr) with wavy flow occurring in some of the tests or during periods. The wavy flow that was observed was of a notably mild and pattern type. This would indicate that a certain portion of the tube evaporator was completely dry throughout a run, and that the part of the tube that was wet was of greater area for Series B than Series A, if compared at the same input.

At higher flow rates, extreme variations in type of flow were noted. In Series C (120 lb/hr) two definite types were noted, one of which was the normal wavy type and the other was of an irregular wavy flow that had a tendency to slug. The wavy flow noted here produced a higher wave formation than those observed for Series B. In Series D highly erratic wavy flow with slight slugging was observed. In Series E the entering flow nearly filled the tube.

The effect of flow rate for simultaneously flowing and boiling Refrigerant-22 is noticeable, especially at lower heat flux densities. This can be observed from Fig. 7.

At low heat flux densities the turbulence due to bubble motion is correspondingly small, and in this case the turbulence produced by fluid flow predominates. Also the area of the tube in contact with the liquid is increased with higher rates of fluid flow. This is due primarily to the increased turbulence of the fluid flow and to a lesser degree by the raising of the liquid level with increasing flow. Accordingly the temperature difference required to transfer a given amount of heat is decreased with an increase of flow rate especially when the heat flux density is low.

As the heat flux density is increased, the bubbling turbulence becomes more violent, until the tube is apparently completely wet. A change in the flow rate in this case does not alter the wetted area. The effect of fluid turbulence is reduced noticeably for this case as an increase in flow rate causes a much smaller decrease in temperature difference. At extremely high heat flux densities where the bubbling turbulence is extremely violent, the effect of fluid turbulence is completely eliminated, and the correlation approaches that of pool boiling.

A plot of the experimental data of Baker and Miller[17] for Refrigerant-11 and data for Refrigerant-12 are shown in Fig. 9 along with the plot of the present data.

The break in the slope of the curves for Refrigerant-11 were not fully explained. A few points observed during the present experiments indicate that a similar break may occur for Refrigerant-22 at low heat flux densities.

Flow rate appears more predominately in the correlation for Refrigerant-11 than for Refrigerant-22 for flows greater than 120 lb/hr but less for flow rates less than 120 lb/hr.

The break that occurred in Equation (5) at a flow rate between 90 and 120 lb/hr supports similar breaks observed for Refrigerant-11, Refrigerant-12, and methyl chloride. It is significant that the break occurs at approximately the same flow rate which may indicate it is independent of the refrigerant. It is believed that this break occurs because of the change in the fluid flow from a stratified flow to an irregular flow in this range. Additional points would be needed for all refrigerants with various evaporator tube diameters to accurately determine the exact point of break before a definite conclusion could be reached.

Rohsenow recommended for pool boiling that the exponent of the first term on the right side of Equation (4) be $\frac{1}{3}$; however, in the light of experiments by Baker,[14,15,17] Miller[16] and the present tests, the possibility exists that it varies for different refrigerants and flow rates.

As mentioned previously, Baker used a Reynolds number term, for correlating Refrigerant-12 data and did not have a variable exponent except that he used $\frac{1}{3}$ for flow rates below 120 lb/hr and $\frac{3}{4}$ for those greater than 120. It was noted that in some of the runs below 120 lb/hr the deviations encountered suggested a higher power than $\frac{1}{3}$. The break in the data at high temperature difference for Refrigerant-12 was not apparent in this experiment for Refrigerant-22, however, these higher temperature differences were not required for Refrigerant-22 for the same heat flux density.

A comparison for Refrigerants-11, -12 and -22 is depicted by Fig. 9. A vertical displacement of the correlating lines is expected because of the unavailability of adequate information to define the surface-fluid combination. From Rohsenow's combination, the same slope should be expected for all refrigerants and this should also be independent of the flow rate. The flow characteristics of the two phase system is believed responsible for the observed behavior.

The authors are of the opinion that independent studies of the boiling and flow properties are necessary to gain complete information to describe the basic behavior. The independent studies are currently active at the University of Kentucky. One study involves a rotating cylindrical evaporator immersed in various liquid refrigerants. Wetting is uniform and complete. A second study involves the analysis of a co-current liquid-vapor phase heat exchanger without boiling. A reasonable expectation would be the feasibility of superimposing data from these independent studies to establish the behavior of a simultaneously flowing and boiling system.

CONCLUSIONS

The objective of this research has been fulfilled in that experimental heat transfer data for simultaneously flowing and boiling Refrigerant-22 were obtained and these data were correlated satisfactorily in terms of the physical properties of the refrigerant and the thermodynamic characteristics of the system.

The variable exponent of the correlating relationship, Equation (8), is believed to be necessary because of (1) flow turbulence and (2) the heat flux density based on the total inside area.

The effect of the fluid flow rate is significant in correlating simultaneously flowing and boiling Refrigerant-22 inside horizontal tubes.

The turbulence created by fluid flow is noticeably important when the bubbling turbulence is low as is the case for low heat flux densities. When the bubbling turbulence is violent, as is the case for high heat flux densities, the influence of flow turbulence is completely obliterated.

The portion of the tube area in contact with the liquid increases with higher rates of fluid flow, especially at low heat flux densities. At higher heat flux densities, the tube appears completely wet from the bubbling action, and consequently changes in rates of flow do not significantly alter the degree of wetting.

The temperature difference required for the transfer of a given heat input at low rates is materially reduced with increasing flow rates. The effect of flow rate is decreased with increasing heat transfer rates.

The characteristics and effects of fluid turbulence and wetted area are apparently interdependent. A separate investigation of each should yield a better understanding of the basic parameters.

The inclusion of the subcooled data points caused deviations slightly greater than those

(*Continued on page* 110)

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Others are saying—

that . . . . . although mechanical refrigeration has long been the standard in low-temperature systems for environmental test chambers, liquid carbon dioxide is now being used successfully as a chamber coolant for special applications. *Test Engineering, February 1959, p 6.*

that . . . . . here is presented evidence in favor of a vapor-liquid exchange mechanism with an explanation of the insensitivity of boiling heat flux to the level of subcooling. The method presented allows the deductions of correlations for nucleate boiling which give the dependence of heat flux on superheat and system pressure. *Journal of Heat Transfer, February 1959, p 43.*

that . . . . . drying with superheated steam-air mixtures could be applied to continuous drying equipment far more broadly were the design of facilities given greater consideration. Under the same conditions of mass flow and temperature, superheated steam gives a faster drying rate during the constant rate period than does air. *Industrial and Engineering Chemistry, March 1959, p 275.*

that . . . . . it can be expected air conditioning will be applied to passenger liners on an ever-increasing scale; in larger vessels the electrically driven plant will be superseded by turbine - driven centrifugal compressors; the use of plastics ducting to eliminate the need for duct insulation may well develop; and, condensate may be used for boiler feed or drinking purposes after treatment. *The Journal of Refrigeration, January/February 1959 p 2.*

that . . . . . sound measurements in school buildings show the need for greater use of acoustical materials to minimize room, corridor, external and duct noises distracting to pupils. It is concluded that there is a need for further investigation in the psychological conditions of the sound environment. *Texas Engineering Experiment Station News, December 1958, p 16.*

that . . . . . research conducted at the National College for Heating, Ventilating, Refrigeration and Fan Engineering upon a fan noise attenuator for high-velocity air conditioning systems shows the governing factor for the amount of noise reduction required will normally be that the noise of the air conditioning system may not be permitted to obtrude over the background noise of the conditioned space. *Journal of The Institution of Heating and Ventilating Engineers, February 1959, p 299.*

that . . . . . curtains of air have many proven and future probable uses including open air restaurants, solariums, hothouses, enclosed gardens, swimming pools. Horizontally-jetted air streams are specifically proposed. *Consulting Engineer, March 1959, p 108.*

that . . . . . a study reported to The Institute of Refrigeration represents an endeavor to provide an economic evaluation of large ammonia absorption refrigerating machines in the evaporator temperature range of —20 to 60 F. *The Journal of Refrigeration, January/February 1959, p 15.*

that . . . . . a report of the Joint Committee of the American Society of Heating and Air Conditioning Engineers and of the American Institute of Architects reviews the economics of air conditioning upon the basis of a service to the engineering and architectural professions. *Journal of the AIA, February 1959, p 57.*

that . . . . . frozen food packs in polyethylene bags, ranging in capacity from 20 oz to 2 lb, offer a new market and appeal to thrifty buyers with home freezers and large families, who purchase with economy. *Canner/Packer, March 1959, p 28.*

People

Meetings ahead

April 1-3—Gas Appliance Manufacturers Association, Annual Meeting, American Hotel, Bal Harbour, Fla.

April 2-5—The Refrigeration Research Foundation, Statler and Mayflower Hotels, Washington, D. C.

April 5-9—National Association of Refrigerated Warehouses, Statler and Mayflower Hotels, Washington, D. C.

April 5-9—Edison Electric Institute, Annual Convention, New Orleans, La.

April 29-May 4—Oil Heat Institute of America, Inc., Annual Convention, Seattle, Wash.

May 3-6—Air-Conditioning and Refrigeration Institute, Board and Annual Meeting, The Homestead, Hot Springs, Va.

May 17-20—Industrial Heating Equipment Association, Inc., The Homestead, Hot Springs, Va.

May 25-28—Design Engineering Show, 4th Annual Exposition and Conference, sponsored by the machine design division of American Society of Mechanical Engineers, Philadelphia, Pa.

June 11-13—Heat Transfer and Fluid Mechanics Institute, University of California, Los Angeles, Calif.

June 22-24—American Society of Heating, Refrigerating and Air-Conditioning Engineers, annual meeting, Lake Placid, N. Y.

August 19-26—10th International Congress of Refrigeration, Copenhagen, Denmark.

October 5-7—American Gas Association, Annual Convention, Chicago, Ill.

October 30-November 2—Refrigeration Service Engineers Society, Annual Convention, Atlantic City, N. J.

November 2-5—11th Exposition of the Air-Conditioning and Refrigeration Industry, Atlantic City, N. J.

February 1-4—American Society of Heating, Refrigerating and Air-Conditioning Engineers, Semi-Annual Meeting, Dallas, Texas.

Franklin Y. Carter, second treasurer of ASHRAE, has been named manager, market planning, of the Detroit Controls Div of American Standard. A graduate of the University of Kentucky (BSME, 1931) he has served on several former ASRE committees, and has authored many papers presented before local and national meetings. In his previous position with Detroit Controls he was product planning manager, air conditioning controls.

John N. Kelley retired last month as manager of fruit transportation for Fruit Dispatch Company and as director of transportation and general research of the parent United Fruit Company. He was presented the William Garfitt Foundation Award in 1947 for his contributions in the field of the transportation of perishables. He is a recently appointed member of the National Academy of Science's U. S. committee for the IIR.

Gustav E. Olsen, Vice President and Manager of Fitsgibbons Boiler Company, New York, died recently. Honored by his alma mater, Brooklyn Polytechnic Institute, several years ago as an outstanding alumnus, he has also served as president of the New York Chapter of the former ASHAE.

David J. Wood becomes associated with Frick Company as Assistant Sales Manager. He will devote most of his time to the resale of equipment through franchised connections. He is a graduate of Fenn College, Cleveland, Ohio.

Evan A. Joy, deceased, was with the Los Angeles Board of Education as associate mechanical engineer. He worked in the design of school buildings.

John S. Burke, a member of the dealer sales div of New Orleans Public Service, Inc., died recently. A past chairman of the New Orleans section of the former ASRE, he was instrumental in organizing the Delta Chapter of the former ASHAE, and in 1947 served as its president.

Vernon C. Dean, Consulting Engineer, Grand Rapids, Mich., died at the beginning of the year.

Jay H. Jarret, former refrigeration sales manager in Ansul Chemical Company's mid-continent region, has been named assistant to the sales vice president. In this capacity he will be in charge of three sales staff departments. **G. G. Partch** will cover Peoria, Ill., and southeastern Iowa in his new position as sales engineer with Acme Industries.

Joshua L. Garvin, deceased, was owner of Garvin Industries, Charleston Heights, S. C.

George W. Shepherd, Jr., becomes manager of the New Orleans sales office of Trane Company, where he has been sales engineer since 1951. He is a mechanical engineering graduate of North Carolina State University.

A. I. Brown, professor emeritus of mechanical engineering, Ohio State University, died recently. A 1912 graduate of Ohio State, he joined the teaching staff in 1913 and was appointed chairman of the department of mechanical engineering in 1946. His work in the heating, air conditioning and ventilating field was in the research and development of equipment.

Albert A. Giannini, Carrier Corporation, is now Assistant Regional Manager in Northeastern United States, formerly New York District Sales Manager for the Machinery and Systems Div. **August Contardi,** the division's senior salesman in the New York area, will replace him as the New York District Manager. Mr. Giannini is a past president of the New York Chapter of the former ASHAE. Mr. Contardi is vice president, Mechanical Contractors Association of New York.

Harvey F. Swenson, president of the Sweden Freezer Manufacturing Company, has been named to the United States Trade Mission to Calcutta, India, by the federal bureau of Foreign Commerce. One of Mr. Swenson's projects was the development of the soft-serve ice cream dispenser, in the 1930's. The team mission to Calcutta is the second in a series of four in connection with United States Exhibits in India over a two-year span. It is from mid-February to mid-April.

Charles A. Hathaway is now Director of Engineering, Air Impeller Div, Torrington Manufacturing Company. As Chief Engineer with that firm for the last seven years, he supervised the expansion of the Engineering Department. He is a graduate of the Massachusetts Institute of Technology (ME degree) and has done work in research and development.

W. R. Woolrich, consulting dean in the college of engineering, University of Texas, leaves the United States for a year of service as the consultant president of the Middle East Technical University, at Ankara, Turkey. Dean Woolrich, recently elected a life member of the former ASRE, has served in many editorial and technical capacities with the Society.

Arthur W. Edwards was a partner in the Edwards Air Conditioning Sales Company, Cincinnati, before his recent death. Prior to this he had been district manager with the Trane Company. In the former ASHAE he served as Cincinnati Chapter President in 1946-47, and as a member of Council from 1953-56.

Allen B. Harvey will cover the eastern seaboard for All-State Welding Alloys Company, Inc., in his recent appointment with that firm. He was previously with Kerotest Manufacturing Company, and with Easton Chemicals, Inc., as eastern sales manager for refrigerants.

H. W. Meinholtz, representative for several major manufacturers, died in January. He was a past treasurer and past president of the Northeastern Oklahoma Chapter of the former ASHAE and was on the board of governors from May 1957 to May 1958.

New officers of the Air Filter Institute, elected at its annual meeting in Philadelphia recently, include **Edwin F. Snyder,** Manager, Product-Application Engineering, Minneapolis-Honeywell Regulator Company, as first vice president; **W. B. Watterson,** vice president, Air Maze Corporation, chosen second vice president; and **Arthur Nutting,** chief engineer, American Air Filter Company, who becomes secretary-treasurer.

Robert A. Miller, deceased, was a retired executive of Pittsburgh Plate Glass Company. A past president of the former ASHAE Pittsburgh Chapter (1939) he was the author of numerous technical books and articles and served on several Technical Advisory Committees.

Vincent J. Hewitt becomes manager of the equipment, dealer and jobber service departments of Fruit Dispatch Company, banana sales subsidiary of United Fruit Company.

Myron C. Beman, deceased, was a partner in the engineering firm of Beman & Standeven, Buffalo. A life member of the former ASHAE, he was president of the Western New York Chapter in 1932 and a member of the Society's Council from 1934-39.

Ray E. Lucey, vice president, engineering, Trane Company, retired last month.

Jack M. Strauss, formerly sales engineer in the automotive air conditioning department, is now product planning manager, air conditioning and refrigeration controls, Detroit Controls Div, American Standard.

BULLETINS

Deoxidizer. For removing oxygen from boiler feed water, Santosite is the subject of a revised technical bulletin. Application and effectiveness factors are reviewed for this anhydrous sodium sulfite.
Monsanto Chemical Company, Inorganic Chemicals Div, St. Louis 24, Mo.

Fan and Limit Controls. Three lines, in several types each, of fan and limit controls are described and illustrated in 4-page Folder R-1630.
White-Rodgers Company, 1209 Cass Avenue, St. Louis 6, Mo.

Sprayed Coil Dehumidifiers. Including not only descriptive and specification details but also application recommendations, Bulletin 9427 is a 24-page engineering guide to draw-through and blow-through sprayed coil dehumidifiers of this manufacturer. There are 9 pages of specific data useful toward the solution of related air-handling problems.
American-Standard, Industrial Div, Detroit 32, Mich.

Control Selector. For use with single and polyphase, 6 cycle, 1800 rpm, 40 C, ac squirrel cage motors, this wall chart type of motor control selector is intended as first aid to specifier.
Arrow-Hart & Hegeman Company, Hartford, Conn.

Variable-speed Pulleys. Designed to preclude freezing and sticking, there are five sheaves serving for 2, 3, 5, 10 and 15 hp whose features are delineated in Bulletin 4101, 12 pages. Power is transmitted from the stationary flange through a removable sleeve cap, keyed to the moving flange by torsionally resilient rubber keys.
T. B. Wood's Sons Company, Chambersburg, Pa.

Registers and Grilles. Within 20-page Catalog 59 C.R. appear specification, performance, price and application details for a full line of registers and grilles.
Air Control Products, Inc., Coopersville, Mich.

Electronic Air Cleaners. Of the high velocity type, the Electro-Cell electronic air cleaners described and illustrated in 16-page Bulletin 258A are shallower, smaller in face area and require less space than have predecessor precipitators. As cited herein, power requirements are low; maintenance is automatic. These units are

(Continued on page 114)

What ASHRAE Chapters are doing

Refrigerating equipment design and its applications are in a new phase, President Boling noted before one chapter last month. Elsewhere, several speakers emphasized the many types of equipment available for both heating and cooling comfort. Vibration and moisture problems, thermoelectric refrigeration and noise control gained attention.

CLEVELAND . . . Engineers of the refrigeration and air conditioning industries are now engaged in a race to find the best materials to use in applying thermoelectric refrigeration, said Jim K. Nelson, experimental engineer for Westinghouse Electric Corporation. These materials must be of the semi-conductor type, he added, at the March meeting of this group.

Difficulty in achieving proper coefficient performance will make large refrigeration and air conditioning applications a problem for the industry, he concluded.

WISCONSIN (H) . . . John Engalitcheff, Jr., President, Baltimore Aircoil company, met with these chapter men in mid-March to give his views on water conservation equipment, such as cooling towers, evaporative condensers and the like.

CINCINNATI . . . Use of Air Cooled Condensers for Air Conditioning was presented at the February meeting, a joint one, by Jack Smith, Carrier Corporation.

PHILADELPHIA (R) . . . Giving a brief history of the development of the convection condenser within his company, Ray Edwards, Edwards Engineering, presented his paper on the Development of Large Tonnage Convection Refrigerant Condensers, in early February.

He explained the basic engineering theory for convection condensers, and application considerations, such as ambient effect, stack effect, sun effect and wind effect.

It would appear, he noted, that this type of condensing system has much in its favor concerning first cost, maintenance and operating cost. It has a built-in overload protection, he pointed out, so that as the load increases, the temperature differential increases only slightly to balance the greater heat dissipation. The larger the area of the condenser, he said, the more efficient the performance.

At a pre-dinner educational forum conducted by Michael Pelosi, Chief Engineer, Bryant Air Conditioning Corporation, Methods of Determining Dehumidified cfm in Cooling Load Calculations were presented.

NORTHEASTERN OKLAHOMA (H) . . . Showing the technical formulas applying to vibration and discussing the several isolating mediums, their advantages and limitations, Jack Harris, executive engineer, Korfund Company, addressed this group in February on vibration problems in equipment and control and isolation of it.

LOUISVILLE (R) . . . Most manufacturers are leaning toward package equipment including controls and wiring, R. E. Olson said at the February meeting. The Manager of Residential Heating and Cooling Systems, American Standard, discussed the important areas of compressor design and life. He included equipment mounting, service and estimating service warranty costs in his presentation.

WESTERN MASSACHUSETTS (H) . . . Comfort control in educational institutions was the main topic in February. Frank Stanton of the School Air Systems Div of American Air Filter Company spoke on Air Conditioning for Schools is Coming.

NEW YORK (R) . . . ASHRAE President Cecil Boling, in addressing last month's meeting, noted that changes in the refrigeration and the air conditioning branches of the new society's activities have included shifts in equipment design emphasis and in the marketing of components and complete assemblies. He covered current and future trends in the industry in general.

ILLINOIS (H) . . . Lower temperature air in a larger volume is more satisfactory for heating than a higher temperature air of smaller volume, Dr. Thomas Bedford of London, said at the February meeting. He based his conclusion on a survey of floor radiant panel installations in school rooms of Derbyshire, England. There it was noted that the temperature of floor panels should be less than 80 F.

Also giving his observations on personal experience with effective temperature problems of British

CHAPTER AND SECTION DATES*

	Apr.	May	June
Atlanta (H)	13	11	—
Austin (H)	16	21	—
Arizona (H)	6	4	1
Arkansas (H)	21	19	—
Baltimore (H)	15	20	—
Baltimore (R)	2	7	—
Baton Rouge (H)	15	20	—
Birmingham (R)	18	14	11
Bluegrass (H)	13	11	—
British Columbia (H)	8	13	—
Boston (R)	7	5	—
Buffalo (R)	—	7	4
Carolina (R)	10	8	—
Central Michigan (R)	16	21	18
Central New York (H)	15	20	—
Central New York (R)	15	13	11
Central Ohio (H)	20	18	—
Central Pennsylvania (R)	8	6	12
Chicago (R)	9	14	—
Cincinnati (H)	7	5	—
Cincinnati (R)	20	—	—
Cleveland (R)	4	2	—
Colorado (R)	13	11	—
Columbus (R)	27	25	—
Connecticut (H)	16	21	18
Connecticut Valley (R)	9	14	5
Dallas-Fort Worth (R)	13	11	—
Dayton (R)	14	23	—
Delta (H)	21	19	—
Detroit (R)	6	4	—
El Paso (H)	20	18	—
Empire State Capital (H)	8	13	—
Erie (R)	14	10	9
Evansville (R)	7	5	5
Florida West Coast (R)	8	13	10
Fort Worth (H)	15	20	—
Georgia (R)	13	11	8
Golden Gate (H)	2	7	—
Hudson-Mohawk (R)	20	18	—
Houston (R)	17	15	—
Illinois (H)	13	11	—
Illinois-Iowa (H)	20	18	—
Indiana (H)	28	26	—
Indianapolis (R)	21	19	20
Inland Empire (H)	6	4	—
Iowa (H)	14	11	—
Jacksonville (H)	7	5	2
Johnstown (H)	—	—	—
Kansas (H)	7	5	—
Kansas City (H)	6	4	—
Kansas City (R)	20	18	15
Long Island (H)	13	11	—
Los Angeles (R)	20	18	15
Louisville (R)	27	11	—
Manitoba (H)	23	28	—
Massachusetts (H)	21	19	—
Memphis (H)	20	18	15

* Pending establishment of unified ASHRAE chapters, meetings of former ASHAE Chapters (H) and former ASRE Sections (R) are proposed upon the basis indicated above for early months.

naval vessels and other English industrial uses, the speaker referred to equivalent heat, which considers dry bulb and wet bulb temperatures, in addition to air movement and radiation. Radiation conditions are an important factor in the consideration of good equivalent temperature, he emphasized.

He noted that the difference in average satisfactory temperatures in this country is several degrees more than that normally considered adequate in Europe.

SOUTH FLORIDA (R) . . . Fan performance was demonstrated and followed by a discussion of the various differences in forward curved and backward curved blades at the February meeting of this chapter. A. P. Rosenquest, Chief Engineer, Chicago Blower Corporation was the guest speaker.

GOLDEN GATE (H) . . . Is Plumbing Necessary? Jerry Kasin asked at the February meeting of this chapter, and then answered with a survey of the history of sanitation and plumbing beginning with the Romans. A brief look at the plumbing of the future was included.

Meeting in March, members planned to hear Ray W. Hawksley, consulting chemical engineer, on Scale and Corrosion in Closed Heating and Air Conditioning Systems and Cooling Towers.

OKLAHOMA (H) . . . Statistics concerning water, its prevalence and properties, were revealed by R. V. Blanche, regional manager, National Aluminate Company, at the February meeting.

Assisted by Edward Oekle, district manager of the same firm, Mr. Blanche discussed briefly the use of chemical treatment in reducing corrosion and scaling in water systems. He emphasized that proper mechanical techniques such as blow-down, deaeration and the like are essential, even with modern chemical treatment techniques. His lecture was entitled Water—A Unique Engineering Material.

BUFFALO (R) . . . Moisture Incorporated was the March topic here. Charles C. Grote, District Manager of Sporlan Valve Company reported on the solution to problems of moisture, acid and dirt in a refrigeration system.

SOUTHERN CALIFORNIA (H) . . . Two speakers appeared before this chapter in February. Robert Prock demonstrated a mechanism which measures amplitude and frequency of vibration. It determines the quantity of correction required and where such correction must be applied.

Noise criteria and typical causes of noise production in fans were the subject of the second speaker, Ken Eldred, who covered noise transmission of ventilation systems.

DALLAS-FORT WORTH (R) . . . Sporlan Valve Company Manufacturing Sales Manager, Jack Dannels, accompanied his address on Moisture – Its Effect on Air Conditioning and Refrigeration Equipment with color slides, at the February meeting.

BRITISH COLUMBIA-WESTERN CANADA . . . Meeting jointly, members of both local societies heard Walter Featherstone as he demonstrated safety devices in flame control equipment. He included an outline of the history of such equipment for use in heavy fuel oils and natural gas, covering flame rods, photo tubes, lead sulphite cells and sound frequency equipment.

LONG ISLAND (H) . . . Problems of Noise Control in Air Conditioning Systems were covered in February when Charles N. Rink, Manager Industrial Acoustics Company, met with these chapter members.

PHILADELPHIA (H) . . . Information on Electrical Space Heating—Today and Tomorrow was given recently when Robert Ruzicks, Manager, Residential and Commercial Space Heating, J. V. Calhoun Company, spoke to the group.

ERIE (R) . . . Plastics pipe covering and insulation were the subject of the talk given in February by Stephen MacFarland, district representative of the Armstrong Cork Company.

PITTSBURGH (H) . . . Different patterns and methods of air distribution were explained at the February meeting. Harold E. Straub, research engineer, Titus Manufacturing Corporation spoke on Principles of Room Air Distribution. He reviewed the results of ASHAE research on this subject.

LOS ANGELES (R) . . . Hosting a Region 9 annual meeting in March, this chapter arranged to hear four technical papers. Edward W. Simons covered Developments in the Application of the Air Curtain in Refrigeration and Air Conditioning; M. B. Grier chose the Thermal Electric Heat Pump (Peltier Effect) and its Applications as his subject; The Practical Evaluation of De-

(Continued on page 127)

	Apr.	May	June
Miami Valley (H)	1	6	—
Michigan (H)	13	11	—
Mid South (R)	13	11	—
Milwaukee (R)	7	5	2
Minnesota (H)	6	4	—
Mississippi (H)	27	25	—
Mobile (R)	22	27	—
Montreal (H)	20	18	—
Montreal (R)	13	11	—
National Capital (R)	7	8	—
Nebraska (H)	14	12	9
New Mexico (H)	21	19	16
New Orleans (R)	6	4	—
New York (H)	20	18	—
New York (R)	6	4	6
Northeastern Oklahoma (H)	14	12	—
Northern Alberta (H)	15	20	—
Northern New Jersey (R)	—	—	—
Northern Ohio (H)	13	11	—
Northern Piedmont (H)	—	—	—
North Jersey (H)	14	12	9
North Texas (H)	20	18	—
Oklahoma (H)	13	11	—

	Apr.	May	June
Ontario (H)	6	4	—
Ontario (R)	3	1	5
Oregon (H)	9	7	—
Oregon (R)	30	28	—
Ottawa Valley (H)	21	19	—
Pacific-Northwest (R)	—	—	—
Panama Canal Zone (R)	15	20	17
Philadelphia (H)	9	14	11
Philadelphia (R)	3	8	—
Pittsburgh (H)	20	18	—
Pittsburgh (R)	4	—	—
Puget Sound (H)	14	12	—
La Ville de Quebec (H)	—	—	—
Richmond (R)	6	4	—
Rochester (R)	4	—	—
Rocky Mountain (H)	1	6	—
Sacramento Valley (H)	1	6	3
San Antonio (R)	14	12	9
San Francisco (R)	23	21	—
San Joaquin (R)	21	19	16
Savannah (H)	—	—	—
Shreveport (H)	16	21	—
South Carolina (H)	20	18	—

	Apr.	May	June
Southern Alberta (H)	—	—	—
Southern California (H)	7	4	—
Southern Connecticut (R)	21	19	—
Southern Piedmont (H)	27	25	—
South Florida (R)	14	12	9
South Texas (H)	17	15	19
Southwest Texas (H)	21	19	16
St. Louis (H)	8	13	—
St. Louis (R)	20	18	—
Switzerland (H)	—	—	—
Toledo (H)	6	4	—
Twin City (R)	13	11	5
Utah (H)	16	21	—
Virginia (H)	15	20	—
Washington, D. C. (H)	8	13	—
Western Canada (R)	27	25	22
Western Massachusetts (H)	23	23	—
Western Michigan (H)	13	11	—
Western Michigan (R)	1	6	—
Western New York (H)	13	11	—
West Texas (H)	7	5	—
Wichita (R)	21	19	—
Wisconsin (H)	20	18	—

Candidates for ASHRAE Membership

Following is a list of 215 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before April 30, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted upon by the Board of Directors.

Note: * Advancement † Reinstatement

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BROWN, T. S., Sales Engr., Johnson Service Co., West Hartford.

KAZIMER, G. A., Plt. Mgr., The Bristol Sheet Metal Co., Wallingford.

MOLSTEIN, EDWARD, Exec. Vice-Pres., Connecticut Heat & Fuel Co., New Haven.

Massachusetts

BALLOU, W. R.,* Dir., South Shore School of Refrigeration, Wollaston.

JARDIN, ADOLPH, Designer, Francis Assocs., Marion.

WATSON, G. C., Methods Tech., Fenwal Inc., Ashland.

New Hampshire

COBE, H. H.,* Pres., Cobe & Foster Inc., Manchester.

New Jersey

ACKERMAN, C. W., Pres., Associated Engineers, Inc., Clifton.

BAILEY, F. W., Proj. Engr., Jet Heet Corp., Englewood.

HARDING, T. G., Sales Engr., Climate Control, Prod. Div., Newark.

KOELLHOFFER, D. B., Supvsr., Engr. & Drafting Sec., Blocker Corp., Union.

SCHULSTADT, H. H., Owner, Middletown Refrigeration, Middletown.

SHAFIR, J. S., Engrg. Aide, Worthington Corp., East Orange.

New York

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BOMMELJE, ABRAHAM, Appl. Engr., Delco Appliance Div., General Motors Corp., Rochester.

CASARSA, R. R., Salesman, W. M. Sommerhays Sons, Rochester.

CRIPPS, E. F., Chief Engr., S. J. O'Brien Cos., New York.

CROSSMAN, R. F., Mgr. Export Units Engrg., Carrier Corp., Syracuse.

DI BENEDETTO, ETTORE, Chief Engr., Air & Refrigeration Corp., New York.

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HOOKER, T. F., Equip. Engrg. Cons., Eastman Kodak Co., Rochester.

JOHNSON, R. W. Sr., Dvlpt. Engr., Carrier Corp., Syracuse.

LENTS, J. M., Sales Mgr., Kenmore Machine Products, Inc., Lyons.

LYNCH, JEREMIAH, Watch Engr., N. Y. U. Medical Center, New York.

MEYERS, LAURENCE, Sales Engr., Slant/Fin Radiator Corp., New York.

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PIZER, M. M.,† Mgr., Engrg. Service Dept., Slant/Fin Radiator Corp., New York.

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SCHWIEGER, A. T., Owner, Fordham Air Control Co., New York.

SELLMAN, N. D., Br. Mgr., York Wholesalers, New York.

TOTHERO, D. L., Dvlpt. Engr., Carrier Corp., Syracuse.

Rhode Island

HAUSSLER, C. F., Research Engr., Grinnell Corp., Providence.

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MISKELLY, H. E., Outside Supt., Riggs Distler Co., Baltimore.

PAULUS, J. D., Proj. Engr., Whitman, Requardt & Assocs., Baltimore.

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GARBARINO, A. P., Service & Installation, McDonald A-C, Miami.

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TAYLOR, J. L., Indus. Engr., The Ohio Fuel Gas Co., Toledo.
WELLS, G. D., Sales Engr., Cincinnati A-C. Co., Cincinnati.

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PLOTKE, W. J., Assoc., Anthony Dvorak Co., Chicago.
RAY, R. F.,* Mech. Draftsman, A. Epstein & Sons, Chicago.
SCHAFFER, P. R., Partner, Slaughter, Schaffer & Assocs., Chicago.
YOUNGREN, J. L., Engrg. Dept. Head, Youngren's, Aurora.

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KRONICK, H. H., Partner, Greenfield & Kronick, Detroit.
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Wisconsin

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O'BRIEN, J. B., Southern Dist. Mgr. Surface Combustion Corp., New Orleans.

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HART, H. K., Chief Clerk, Noland Co., Inc., Jackson.

Missouri

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DAWSON, WILLIAM, Sales Engr., Water Cooling Equip. Co., St. Louis.
DREIFUSS, PHILIP, Regl. Mech. Engr., Div. of Hospitals & Medical Facilities, Public Health Service, Kansas City.
FATCHETT, J. R., Appl. Engr., York Corp., St. Louis.
FRANK, L. H.,* Asst. Engr., Burns & McDonnell Engineering Co., Kansas City.
HALL, L. P., Jr., Sales Repr., E. I. du Pont de Nemours & Co., Webster Groves.
KNEPPER, E. R., Field Engr., Alco Valve Co., St. Louis.
MISTER, M. M., Owner, Arrow Refrigeration Co., Hyattsville.
SAHLER, D. F., Sales Engr., Sporlan Valve Co., St. Louis.

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PICHT, D. E., Factory Repr., Dunham-Bush Inc., Dallas.
SCOGGIN, J. V., Jr. Engr., Zumwalt & Vinther, Lubbock.
ZANCKER, I. J.,† Tech., General Electric, Tyler.

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ZINN, R. T., Assoc., Stiles Sales Co., Denver.

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MALLOY, J. C., Master Mech., Armour & Co., Kansas City.

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PETERSON, T. W.,* Br. Mgr. & Sales Engr., Barber-Colman Co., Omaha.
SOLZMAN, I. I.,† Pres., The Gallaher Co., Omaha.
THOMPSON, D. P., Sales Engr., William R. White Co., Inc., Omaha.

REGION X

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RYAN, O. F., Service & Installation Repr., Minneapolis-Honeywell Regulator Co., Tucson.
SECREST, C. D., Combination Engr., Service & Inst. Minneapolis-Honeywell Regulator Co., Phoenix.
SCRUGGS, J. D., Estimator & Mgr., Copeland Sheet Metal, Phoenix.

California

BRINER, C. F., Proj. Engr., American Electronics Inc., El Monte.
ESCHER, W. F., Partner, Swickard & Escher, Glendale.
FADER, E. J., Proj. Engr., Trade Wind Motorfans, Inc., Rivera.
GALBRAITH, R. R., Asst. Prof., California State Polytechnic College, San Luis Obispo.
GRAHAM, R. C., Br. Mgr., Dust Controls, Inc., San Diego.
KOEHLER, F. M.,† Sales, L. B. Marsh Allied Inc., Long Beach.
PHILLIPS, D. E., Engrg. Draftsman, Kohlenberger Engrg. Corp., Fullerton.
SECORD, I. B., Sales Engr., Drayer Hanson Div. National U. S. Radiator, Los Angeles.
STUMPF, C. C., Territory Mgr., Lennox Industries Inc., Los Angeles.
WILLIE, U. T., Engr., Indenco Engineers, Inc., San Leandro.

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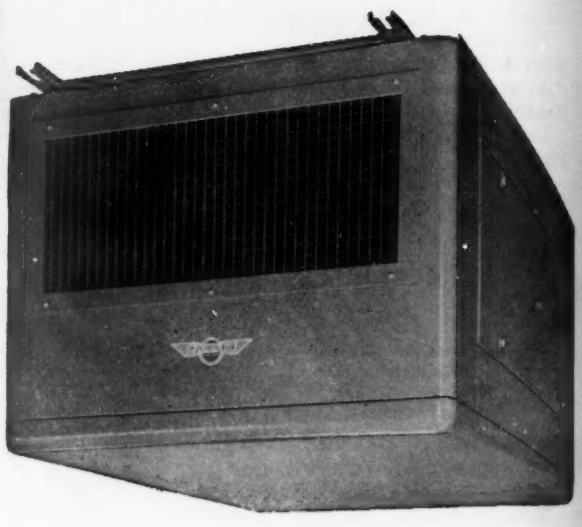
GROSSENBACHER, W. A., Sales Engr., Pacific Pumping Co., Portland.

(*Continued on page* 118)

Keyed to Profit!

LARKIN

Year 'Round AIR CONDITIONER



5 Models — 2 to 10 Tons

COOLS OR HEATS - DEHUMIDIFIES - FILTERS - CIRCULATES AIR

Ideal for:

STORES
OFFICES
BAKERIES
RESTAURANTS
BARS AND GRILLS
CLINICS
FLORISTS
MEAT CUTTING ROOMS

LOW INITIAL COST!

Purchase price is moderate for a unit with this high quality construction, performance, endurance.

EASY TO INSTALL!

Time is money in your business. The LARKIN Air Conditioner is engineered to make installation simple.

EASY TO SERVICE!

Replacing filters from the bottom is simple. Side panels easily removed for services of internal parts.

FACTS AND FEATURES GIVE PROOF

- Larkin air-conditioning coil—eight fins per inch, continuous fin, staggered tubes, for highest efficiency, lowest operating cost
- Heating coils for use with steam or hot water
- Compact cabinet, all-steel, rust-resistant, beautifully finished with baked-on enamel
- Fiberglas insulation
- Pressure-type, centrifugal, dynamically balanced, forward-curved fan wheels
- Self-aligning bearings
- Resilent base motors on adjustable mounts
- Two-direction, adjustable discharge grille
- Easily removed, throw-away filters
- Slotted hanger bars
- Easily installed, easy to service
- Backed by the engineering skill and manufacturing reputation of Larkin Coils, Inc., one of America's oldest makers of commercial and industrial refrigeration and air-conditioning equipment

CAPACITIES AND SPECIFICATIONS (Cooling Coil)

DIRECT EXPANSION RATINGS 80° D. B. 67° W. B. Ent. Air, 40° Ref. Leaving Air					*Dimensions*		
Model No.	*CFM*	*Tons*	*D.B.*	*W.B.*	*Height*	*Width*	*Depth*
AH-2	800	2.06	61.8	57.4	22¾	30	46⅞
AH-3	1200	3.27	60.7	56.8	22¾	35	46⅞
AH-5	2000	5.39	61.0	56.9	22¾	50	46⅞
AH-8	3000	8.16	60.7	56.8	25¾	61	46⅞
AH-10	4000	10.88	60.7	56.8	31¾	61	46⅞

WATCHDOG OF THE NATION'S FOOD AND COMFORT ☆LARKIN☆

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ATLANTA 1, GEORGIA

NEW COMPLETE LINE OF HONEYWELL ACTUATORS AND DAMPERS FOR YOUR EVERY NEED!

Honeywell's new M630 Actuator motor.

Honeywell's D42 Damper

Honeywell Actuators—to position dampers, valves, pre-rotation vanes, skylights, defrosting doors—any application up to 750 pounds of torque

Honeywell can help you satisfy an almost unlimited number of power positioning requirements. You can choose from 11 models which offer: torque ratings from 25 to 750 inch pounds; timing ranges of 7.5 seconds to 4 minutes; 3 control methods—two position, proportioning and floating; 2 drive types—reversible with or without spring return and/or unidirectional.

Other outstanding features of Honeywell's Actuator line include: choice of tandem or sequence switching; 360° adjustable crank arm; ease of mounting and adjustment; quiet, oil-geared operation; sturdy, durable Honeywell components for years of trouble-free operation.

Honeywell, the biggest name in dampers for heating, ventilating, air conditioning and specialized applications

The custom-built features of Honeywell motorized dampers give them an almost limitless versatility. In addition, they give instant response to controllers along with an assured tight seal. They have extra heavy bracing to assure true alignment and consistent accuracy of positioning. Operation is noiseless—and trouble-free.

The custom-built features of the Honeywell Damper line also mean easier ordering, simplified installation. Get complete and detailed information on Honeywell's complete line of dampers and actuator motors by calling your local Honeywell office. Or write Honeywell, Dept. RE-4-20, Minneapolis 8, Minnesota.

Honeywell

H HONEYWELL *First in Control*

30,000
36,000
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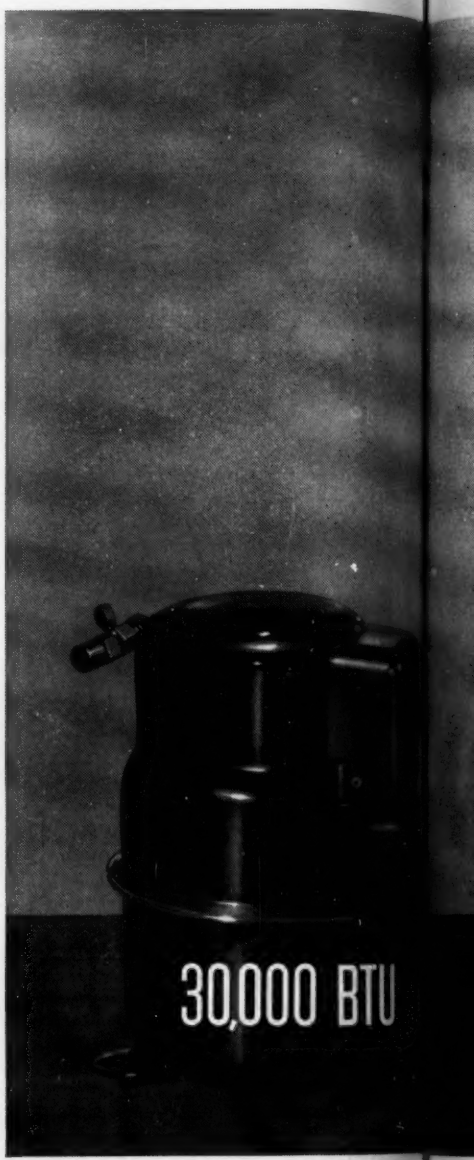
BTU SIZES

SAVE WEIGHT, SAVE SPACE, CUT COSTS!



"Let me tell you a

"As a Bendix-Westinghouse engineer I have just seen the first production models of three new compressors—30,000, 36,000 and 42,000 BTU's respectively—that I am confident will help make history for the air-conditioning industry. No compressors of their size can deliver so many BTU's. No compressors of their BTU ratings were ever so small and compact. And the happy result: Now the industry can design *without compromise* for the home air-conditioning field!

about these three great new compressors"

"For an example of what I mean, look at the air conditioner market for 4-5 room houses. Until now compressors have been either too large or too small. But with the new 30,000, 36,000 and 42,000 BTU rated units you can give this mass market just what it wants.

"Here are some reasons, over and beyond capacities, that help these new compressors surpass anything now on the market.

"One important reason is the surprising savings in space. And this smaller size is due to many factors. Better design is one. The high-speed motors are another because they allow use of smaller pistons with corresponding reductions in the sizes of the crankcases.

"The way vibration is reduced is really important too.

"Prices? I can assure you, as startling as it may seem, that these great new Bendix-Westinghouse compressors are priced lower than any other of similar capacity now on the market. In fact, prices are reduced as much as 25%.

"And they're really compact! Width, for example, is reduced 25%; weight 40%.

"We look upon these compressors as most significant engineering contributions to the industry.

"If you design, manufacture or sell air-conditioning units, you will want further information on how these new compressors, with capacities never before available in such compact size, open up whole new markets for your products."

Bendix-Westinghouse EVANSVILLE, INDIANA

A Division of Bendix-Westinghouse Automotive Air Brake Company, Elyria, Ohio • Export Sales: Bendix International, 205 E. 42nd St., New York 17 N. Y.

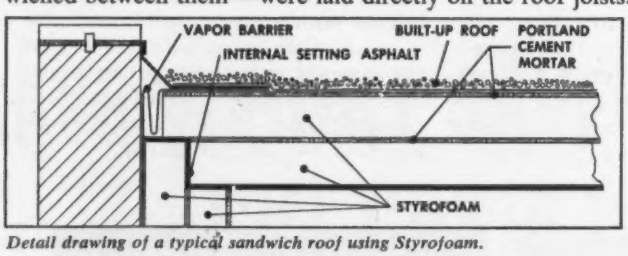
DOW | INSULATION

Designer: Ward Ice Industries, Ft. Smith, Arkansas. *Approved Insulation Contractor: Chris Fiedler Company, Inc., Memphis, Tennessee.*

Unique sandwich roof construction using STYROFOAM® saves $20,000

Rogers Ice and Cold Storage Company, Rogers, Arkansas, put the rigidity, light weight and high compressive strength of Styrofoam* to good use in their new 24,000 sq. ft. cold storage plant.

In this sandwich roof construction, two 4" thicknesses of Styrofoam—with a ¼" layer of portland cement mortar sandwiched between them—were laid directly on the roof joists.



Detail drawing of a typical sandwich roof using Styrofoam.

A ½" portland cement mortar grout was put down over this sandwich and a built-up roof applied in the normal manner. The concrete slab, which is standard for this type of roof, was eliminated because Styrofoam is capable of supporting not only its own weight, but also that of the built-up roof and the foot traffic necessary for inspection and maintenance!

Eliminating the concrete slab allowed a reduction in the amount of structural steel used and, of course, an over-all reduction in labor . . . adding up to a $20,000 savings! Styrofoam, used in the floors and walls, will provide superior insulating efficiency and reduced maintenance for Rogers, too. For more information about Styrofoam in cold storage uses, contact your nearest Styrofoam distributor or write to THE DOW CHEMICAL COMPANY, Midland, Michigan, Plastics Sales Department 2220JZ4.

*Dow's registered trademark for its expanded polystyrene

THE DOW CHEMICAL COMPANY • MIDLAND, MICHIGAN

(*Continued from page* 53)

with 47.7 degree incidence angle S = 0.24, for diffuse solar radiation S = 0.22, and r = 0.27. The intensity of the direct solar radiation on the wall 267 × 0.673 = 180 Btu per (hr) (sq ft), the diffuse solar radiation 35 Btu per (hr) (sq ft), and the outside air temperature 95 F. Substituting these values in Equation 2, the shade temperature for the sun-lit section is $t_s = 0.24 \times 180 + 0.22 \times 35 + 0.27 (95 - 75) + 75 = 131.3$ F, and for the section in the shadow, $t_s = 0.22 \times 35 + 0.27 \times (95 - 75) + 75 = 88.1$ F.

RESULTS DISCUSSED

The solar calorimeter investigations were made under winter and summer conditions. The winter testing, with no solar radiation at night, supplied information regarding the effect of glass-shade spacing, shade emissivity, and shade opening on the heat loss resulting from the temperature difference between the indoor and outdoor air. As no solar radiation was present during these tests, the solar absorptance of the shade material and glass were not important. With a fully drawn roller-shade having about ½-in. opening on the edges, the variation of glass to shade spacing from ¾-in. to 5¼-in. had no significant effect on the heat loss. The emissivity of the shade material for long-wave radiation and amount of shade opening influenced the heat loss as shown by the U values in Table III. The U values are lower for the fully drawn aluminum foil shade (i.e., low emissivity) than for the cloth shades. By sealing the edges of the shade to the window frame the circulation of room air between the shade and glass is stopped, and therefore, the heat loss is lower than with a ½-in. space between the edges of the shade and the window frame. A window having a half-drawn roller-shade will have a U value which is the average of the U values for a fully drawn shade and a window with no shade.

The summer heat gains for a glass-shade combination were investigated in the presence of the solar radiation with fully drawn roller-shades having about ½-in. opening on the edges. The variation in glass-shade spacing from ¾ to 3 in. had no effect on the heat gain; but with 5¼-in. spacing the heat gain was about 5 to 10% higher. The data presented in this paper are for a 3-in. glass-shade spacing.

The heat gain due to solar radiation was less with a shade having a low solar absorbing surface (i.e., white) on the glass side than with a high solar absorbing surface (i.e., dark color). The heat gain was higher with a white translucent shade than with a white opaque shade, although the solar absorptance of the former was less. The reason for this was that the solar energy transmitted through the translucent material more than offset the decrease in the convection-radiation component of the heat gain resulting from the lower solar absorption.

Lowering the emissivity of the surface of the shade for long-wave radiation (i.e., aluminum) on the room side reduced the heat gain due to solar radiation. However, lowering the emissivity of the surface of the shade on the glass side increased the heat gain due to solar radiation with a regular plate glass, but decreased it with a heat absorbing glass.

A dark green shade in combination with heat absorbing glass gave a lower heat gain than the same shade when used with regular plate glass. This was due primarily to the high heat absorption of the heat absorbing glass in excluding the initial entry of the solar heat, and to the low reflectivity of the green shade. However, with a white shade, heat gains were less with a regular plate glass than with heat absorbing glass. Here, with the regular plate glass, the white shade was able to reflect an appreciable percentage of the solar energy back to the outside. With the heat absorbing glass, although

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less heat was transmitted directly to the shade, a sizeable portion of that which was reflected was absorbed by the glass, thus increasing the glass temperature.

It is interesting to note that the K values given in Table II are a relative measure of the solar heat gain with different type of glass-shade combinations. The smaller the K value, the smaller is the solar heat gain.

Table VII illustrates the effectiveness of the roller shades in reducing heat gain through windows over the entire period of the day during which the direct sun falls upon the window. It is to be noted that the data in this table are based on Table IV values, which were for a window with no set-back. For a window with a set-back, however, heat exclusion is less than given in Table VII.

The long-wave radiation emitted by the room side surface of the shade affects the comfort of the occupants, and the amount of radiation emitted depends on the shade temperature and emissivity. The shade temperature is high with a dark-colored shade and low with a white shade. With an aluminum shade or a dark green shade with aluminum surface on the room side, shade temperature is also high; but, with such shades long-wave radiation emission will be less because of the low emissivity of the aluminum surface.

An approximate value of the convection-radiation component of heat gain can be calculated for any shade other than those tested by the method described in Appendix B, by using Equation B-7 and by taking the U and U′ values in this equation from the curves in Fig. B-1.

CONCLUSIONS

1. The summer heat gains through a glass-roller-shade combination can be obtained from the data presented in Tables IV and V, for East, South and West orientations and for various hours of the day. For outdoor conditions other than those given in Tables IV and V, the summer heat gains and winter heat losses can be calculated from Equation 1 and Table II.

2. The shade temperatures can be calculated from Equation 2 and Table VI.

3. With an opaque roller-shade, solar heat gains are reduced by lowering the solar absorptance of the surface of the shade on the glass side.

4. Effect of a low emissivity surface for long-wave radiation on the room side of the shade is to reduce the heat gain due to solar radiation.

5. Effect of a low emissivity surface on the glass side of the shade is to increase the solar heat gains with a regular plate glass and decrease it with a heat absorbing glass.

6. With an opaque shade having a white surface on the glass side the solar heat gains are less with regular plate glass than with heat absorbing glass. However, with a dark green shade, heat gains are less

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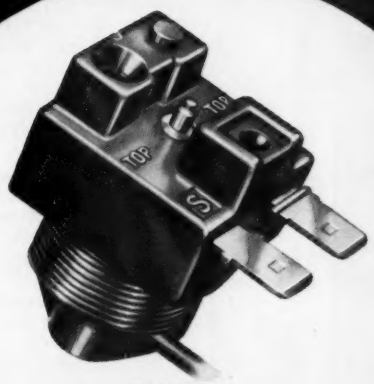

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ACKNOWLEDGMENT

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NOMENCLATURE

C = convection heat exchange, Btu per (hr) (sq ft).
R = radiation heat exchange, Btu per (hr) (sq ft).
CR = convection and radiation heat gain into the room, Btu per (hr) (sq ft).
E = Stefan - Boltzman constant multiplied by the temperature of the surface in Fahrenheit degree absolute to the fourth power, Btu per (hr) (sq ft).
h = Thermal conductance Btu per (hr) (sq ft) (F).
I_v = intensity of solar radiation on a vertical surface, Btu per (hr) (sq ft).
K = a factor, dimensionless.
$\alpha Ig.$ = solar energy absorbed by the glass, Btu per (hr) (sq ft).
αIs = solar energy absorbed by the shade, Btu per (hr) (sq ft).
αIgs = solar energy absorbed by glass-shade combination, Btu per (hr) (sq ft).
t = temperature, F.
t'g = temperature of glass with no solar radiation, F.
Δt = rise in glass temperature due to solar radiation, F.
Q = total heat gain, Btu per (hr) (sq ft).
r = a factor, dimensionless.
S = a factor for determining the roller shade temperature, Btu per (hr) (sq ft) (F).
T = transmittance, dimensionless.
U = overall coefficient of heat transfer, Btu per (hr) (sq ft) (F).
U' = solar heat transfer coefficient, Btu per (hr) (sq ft) (F).
α = absorptance, dimensionless
ξ = emissivity for long-wave radiation, dimensionless.
γ = reflectance, dimensionless.

SUBSCRIPTS

a, g, s — refer to space between glass and shade, glass, and shade.
i, o — refer to inside and outside.

APPENDIX A

Convection-Radiation Heat Gain

Consider a glass and a roller-shade combination with no solar radiation. The heat flow through the window is

$$\text{Heat flow} = U(t_i - t_o) = h_o(t'_g - t_o) \quad \text{(A-1)}$$

where, t'_g is the glass temperature with no solar radiation.

With the solar radiation, however, the solar energy absorbed by the glass and shade increases their temperatures until a balance is reached between the rate of heat absorption by them and the rate of heat dissipation from them by convection and radiation. Assuming a negligible storage, the following energy balance equations can be written for the glass and shade:

$$\alpha Ig = h_o(t_g - t_o) + C_{ga} + R_{gs} \quad \text{(A-2)}$$

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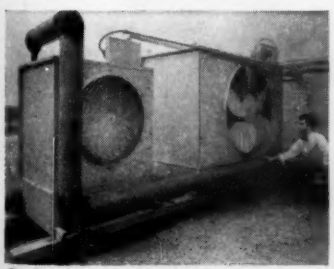

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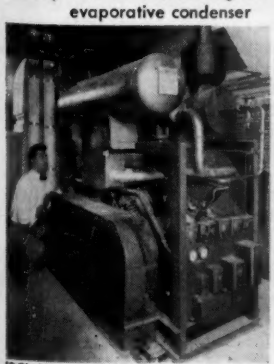

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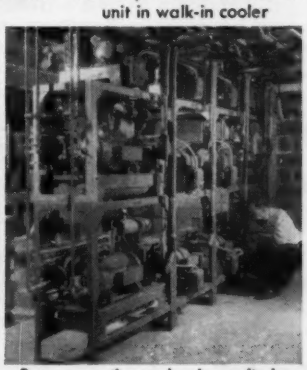

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$\alpha Is = CR_{si} + C_{sa} + R_{sg}$ (A-3)
where, t_g is glass temperature with the solar radiation.
Adding these two equations

$$\alpha Ig + \alpha Is = h_o(t_g - t_o) + (CR_{si} + C_{ga} + C_{sa}) \quad (A\text{-}4)$$

In Equation (A-4), $(\alpha Ig + \alpha Is)$ is the solar energy absorbed by the glass and shade, and $(CR_{si} + C_{ga} + C_{sa})$ is the total convection-radiation heat gain into the room. Let,

$$\alpha Ig + \alpha Is = \alpha Igs \text{ and}$$
$$CR_{si} + C_{ga} + C_{sa} = CR \quad (A\text{-}5)$$

Substituting these in equation (A-4),

$$\alpha Igs = CR + h_o(t_g - t_o) \quad (A\text{-}6)$$

Substituting $tg = t'g + \Delta t$ in equation (A-6)

$$\alpha Igs = CR + h_o(t'_g - t_o) + h_o\Delta t \quad (A\text{-}7)$$

From equations (A-1) and (A-7),

$$CR + U(t_i - t_o) = \alpha Igs - h_o\Delta t \quad (A\text{-}8)$$

In Equation (A-8), Δt is the rise in glass temperature due to the solar radiation; and for a given indoor condition and outside conductance it is a function of αIgs. This suggests that, for a given indoor condition and outside conductance, $CR + U(t_i - t_o)$ should be a function of αIgs.

APPENDIX B

Effect of Outdoor Conductance and Shade Emissivity on the Solar Heat Gain

Consider a combination of a glass and a roller-shade with sealed edges in the presence of the solar radiation. With negligible storage, the energy balance equations for the glass and shade with sealed edges, (i.e. no air circulating from the glass-to-shade spacing into the room) are:

$$\alpha Ig = h_o(t_g - t_o) + h_a(t_g - t_s) \quad (B\text{-}1)$$
$$\alpha Is = CR + h_a(t_s - t_g) \quad (B\text{-}2)$$

From Equations (B-1) and (B-2), eliminating t_g,

$$\left(\frac{1}{h_o} + \frac{1}{h_a}\right) CR + t_s = \frac{1}{h_o}\alpha Ig + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha Is + t_o \quad (B\text{-}3)$$

where, CR and t_s are interdependent variables. For the solar calorimeter, the relation between CR and t_s was given in Reference 5. For the design condition, however, this relation, as suggested in Reference 4, was:

$$CR = 0.27\,(t_s - t_f)^{1.25} + \xi(E_s - E_i) \quad (B\text{-}4)$$

For given values of t_i, h_o and h_a, it is possible to plot CR as a function of the composite term on the right hand side of Equation (B-3), (i.e. $\frac{1}{h_o}\alpha Ig + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha Is + t_o$) from the foregoing relations. This was plotted for 75 F indoor temperature and a 3-in. air space between glass and shade, for the different values of h_o, by taking the air space conductances, h_a, the same as those given in the Guide (p. 176, 1958). The curves obtained in this manner were almost straight lines. Hence, the straight line equation for CR with a sealed shade is:

$$CR = U'\left[\frac{1}{h_o}\alpha Ig + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha Is + t_o - 75\right] \quad (B\text{-}5)$$

and for a 75F outdoor temperature,

$$CR = U'\left[\frac{1}{h_o}\alpha Ig + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha Is\right] \quad (B\text{-}6)$$

where, the slope U′ can be defined as the solar heat transfer coefficient.

The convection - radiation heat gains from the test data for the shade with unsealed edges, after being adjusted for a 75 F outdoor temperature, were compared with those obtained from Equation (B-6). For a shade with a given emissivity, the ratio of CR from the test data and from Equation (B-6) was found to be constant within 10 to 15 per cent, regardless of the solar absorptance of the shade material. Therefore, a variation in h_o alters the solar heat gain for the test data in the same proportion as it alters CR defined by Equation (B-6). This relation was used for adjusting the solar heat gains for the desired outside conductance.

The values of U′ for the shades tested with unsealed edges, as obtained from Equation (B-6), are given in Fig. B-1 as a function of the shade emissivity. The U′ values given in this figure, are for an outside conductance of 3 Btu per (hr) (sq ft) (F) and for equal outdoor and indoor air temperatures. The extended range of values, although only approximate, serves to indicate the effect of shade emissivity on the solar heat gain.

With no solar radiation, the heat flow is due to the indoor and outdoor air temperature difference, and the over-all coefficient of heat transfer (i.e. U) is the factor controlling the

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4" - 6V	220	370	480	.....

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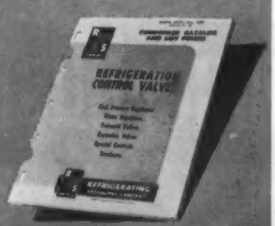

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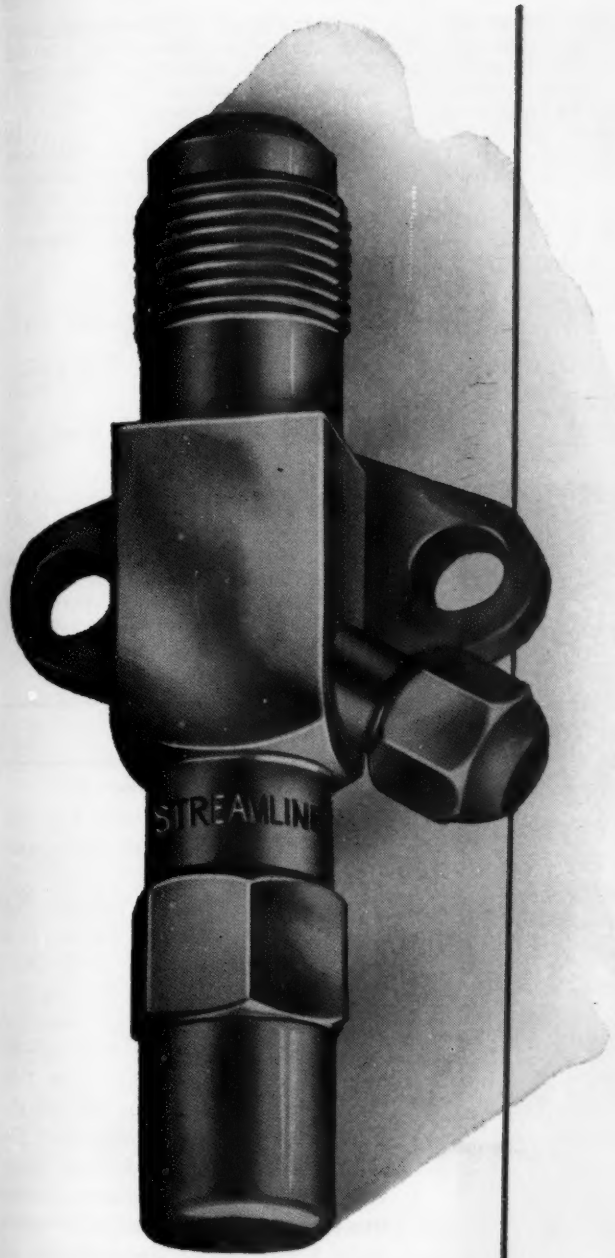

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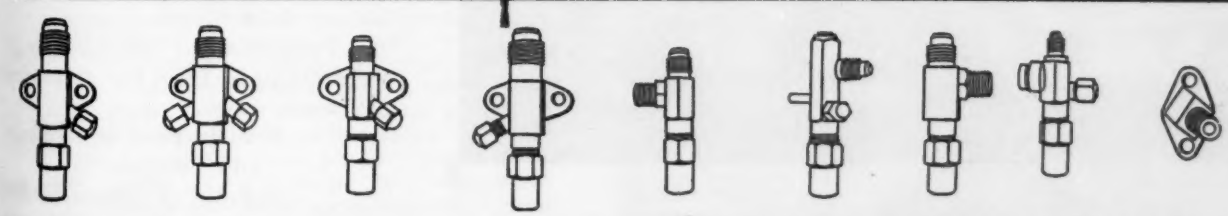

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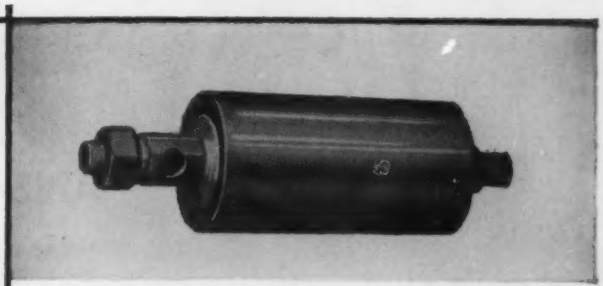

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heat flow. The U values for the different shades tested, based on an outside conductance of 3 Btu per (hr) (sq ft) (F), are also plotted in Fig. B-1.

With the solar radiation and different indoor and outdoor temperatures, the convection-radiation component of the heat gain for any shade with unsealed edges and for an outside conductance of 3 Btu per (hr) (sq ft) (F) can be calculated from the following equation, by taking the appropriate U′ and U values from Fig. (B-1):

$$CR = U'\left[\frac{1}{3}\,\alpha Ig + \left(\frac{1}{3} + \frac{1}{h_s}\right)\alpha Is\right] + U\,(t_o - t_i) \quad \ldots\ldots (B\text{-}7)$$

where, h_s is the conductance of the sealed air space for a 3-in. spacing, and may be taken from the Guide (p. 176, 1958), and t_i is between 70 and 80 F. αIg and αIs can be calculated approximately as follows:

$$\alpha Ig = Iv \cdot \alpha g\ (1 + Tg\ Is) \ldots (B\text{-}8)$$

$$\alpha I_s = I_v \cdot T_g \cdot \alpha s \ldots\ldots\ldots\ldots (B\text{-}9)$$

APPENDIX C

As seen in Fig. 3, for a 75 F indoor temperature and for a given glass-shade combination $CR - U(t_o - 75)$ and αIgs are linearly related as follows:

$$CR - U(t_o - 75) = b \cdot \alpha I_s \quad (C\text{-}1)$$

or

$$CR = b \cdot \alpha Igs + U(t_o - 75) \quad (C\text{-}2)$$

where b = the slope of the curve in Fig. 3.

If the shade is translucent, some solar energy is transmitted into the room, and the amount of this transmitted energy is,

$$\text{Transmitted energy} = T_g \cdot T_s \cdot I_v \quad (C\text{-}3)$$

The total heat gain into the room is the sum of the convection-radiation and transmitted components of the heat gain given by Equations (C-2) and (C-3). Hence, the total heat gain is:

$$Q = b \cdot \alpha Igs + T_g\,T_s\,I_v + U(t_o - 75) \ldots\ldots (C\text{-}4)$$

where, αIgs can be evaluated as follows:

$$\alpha Igs = (\alpha g + T_g\,\alpha_s + T_g\,\zeta_s\,\alpha_g)\,I_v \quad (C\text{-}5)$$

substituting (C-5) in (C-4),

$$Q = [b(\alpha g + T_g\,\alpha_s + T_g\,\zeta_s\,\alpha_g) + T_g\,T_s]\,I_v + U(t_o - 75) \ldots (C\text{-}6)$$

Let

$$K = b(\alpha_g + T_g\,\alpha_s + T_g\,\zeta_s\,\alpha_g) + T_g\,T_s \ldots\ldots (C\text{-}7)$$

Then

$$Q = K.\,I_v + U(t_o - 75) \ldots (C\text{-}8)$$

Where, the K-factor depends on whether the solar radiation is diffuse or direct, the incidence angle of the direct beam, the solar properties of the glass and shade, and an experimentally determined value (b).

ANSUL T-FLO DRIER

The Ansul T-Flo Drier has an insatiable thirst for moisture and acids that can occur in a system. As refrigerant flows through this unique drier, Andrite—Ansul's deep-drying desiccant—greedily adsorbs the harmful intruders. Solids are trapped and held fast, too, so systems stay bright and clean. And with a Super Dry-Eye in the line, the moisture control system—drier, moisture indicator, sight glass— is complete with only 1 installation. The Ansul T-Flo Drier saves valuable time on every job . . . protects the profitability of service contracts, too. The Ansul Line-Flo Drier, with Andrite, is recommended when small standard driers are necessary.

ANSUL CHEMICAL COMPANY
MARINETTE, WISCONSIN

ANSUL

REFRIGERATION PRODUCTS
FIRE FIGHTING EQUIPMENT
INDUSTRIAL CHEMICALS

HEAT TRANSER RATES

(*Continued from page* 89)

for non-subcooled points, however these divergencies are not excessive.

For the range explored, the rate of heat transfer for a given temperature difference from heated horizontal tubes to boiling Refrigerant-22 is greater than for either Refrigerant-11 and/or -12.

The data presented in this paper may be used for design applications but more important, the opinion is expressed that they represent progress toward a better understanding of the basic parameters governing the transfer of heat from a warm surface to a boiling fluid.

Properties of Refrigerant-22 were obtained from the following sources:

E. I. du Pont de Nemours and Company
Organic Chemicals Department
Kinetic Chemicals Div.

(1) Surface Tension of Refrigerant-Compounds (Graph)
(2) Specific Heats of Refrigerant-Compounds (Table)
(3) "Thermodynamic Properties of Freon-22," 1945; "Freon Refrigerants," 1942

A. F. Benning, W. H. Markwood, Jr., "The Viscosities of Freon Refrigerants," REFRIGERATING ENGINEERING, XXXVII (April, 1939), 243.

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ACKNOWLEDGEMENTS — The authors appreciate the interest in and financial support given by the Engineering Experiment Station, the Department of Mechanical Engineering of the University of Kentucky, and The American Society of Refrigerating Engineers. Gratitude is expressed to the Freon Products Div of E. I. du Pont de Nemours and Company for providing refrigerant samples.

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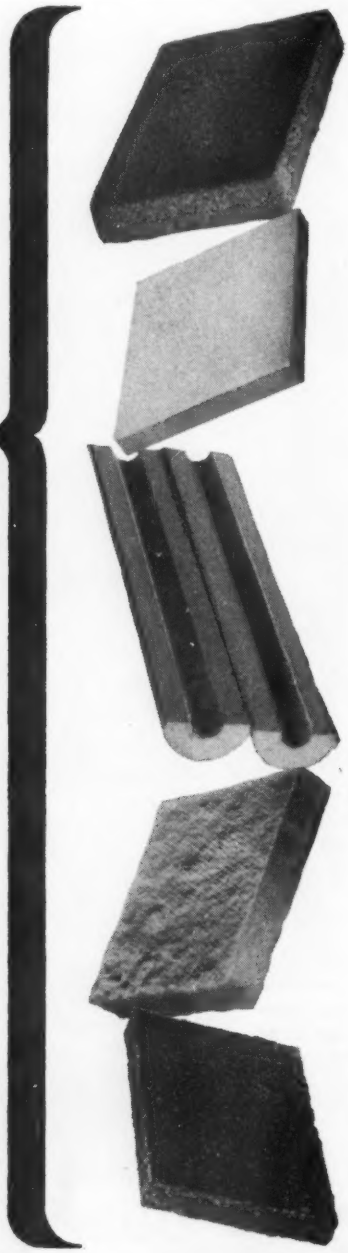

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STANDARDS

(*Continued from page* 82)

and Atmospheric Water Cooling Equipment had ASRE representation and these committees will continue to be represented by ASHRAE representatives.

5 — The **American Society of Electrical Engineers (AIEE)** had ASRE representation on a subcommittee studying the Evaluation of Hermetic Motor Insulation. Such activities will be continued by these representatives as members of ASHRAE.

6 — The **National Association of Corrosion Engineers (NACE)** will have ASHRAE representation on their Intersociety Corrosion Committee. Previously, ASRE was represented on this committee.

7 — The **American Concrete Institute (ACI)** Committee for Residential Concrete Work had ASHAE representation and the Society will continue this activity.

8 — **Industry Association Liaison** —In order that the Society testing standards would complement standards relative to the same subject promulgated by the industry association, ASRE had established with the Air-Conditioning and Refrigeration Institute (ARI) a Standards Liaison Committee. The initiation of similar liaison committees with industry associations in the heating and ventilating fields will be considered by your Standards Com-

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All large capacity pilot piston operated Sporlan solenoid valves are equally applicable to Refrigerants 12, 22, and 500.

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mittee. So that members may become familiar with the areas of intersociety standardization, a listing of all standards and Society representatives will be published in a future issue of the JOURNAL.

COOPERATION WITH GOVERNMENT AGENCIES

A major user of industry standards is the federal government. The Department of Defense has established in many of its subdivisions Standards Departments for the purpose of maintaining liaison with industry groups in the fields of standardization. ASRE had liaison members of project committees from several government agencies and this activity has proven to be useful in the development of military specifications. A number of proposed specifications in areas related to refrigeration and air conditioning have been submitted through this office to project committees of your Standards Committee. Such liaison and review prior to the public dissemination of military specifications is of extreme importance. This activity will undoubtedly increase in the near future due to the expanded scope of ASHRAE.

The theme of the 6th National Conference on Standards in 1955 was Government-Industry Cooperation in Standards. In the keynote address, the Honorable Thomas P. Pike, Assistant Secretary of Defense, pointed out that, "It is wasteful of money and materials to permit our government to operate by one set of standards and industry by another, with no proper connection between the two."

INTERNATIONAL STANDARDIZATION

In foreign countries where their standards are not in agreement with our own, there is an immediate handicap or roadblock to our economic intercourse. A number of our industry standards are of equal importance nationally and internationally.

In order to correlate the activities of various standardization groups throughout the world in the consideration and development of international standards, the ISO (International Standardization Organization) was established. The ISO has become known as the "Vehicle for World Trade." Representation at ISO is by national groups and not as individuals. The U. S. is represented through the ASA. The Society is already actively represented on many of the committees working on international aspects of standardization. Presently the field of refrigeration is most active, however, similar activities in ISO relative to air conditioning and heating will probably be initiated in the future.

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BULLETINS

(*Continued from page* 93)

available in capacities from 1000 to 240,000 cfm and by National Bureau of Standards dust spot test methods indicate an efficiency as high as 97%. **American Air Filter Co., Inc., 215 Central Avenue, Louisville 8, Ky.**

Inner-Fin Heat Exchangers. Solution of the problems of space and weight are emphasized in 8-page Form No. 5003, which relates primarily to the adapting of cooling equipment to the confined systems of electronic equipment, missiles and aircraft. Included are liquid-to-air applications of finned

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radiators and heat exchangers of the oil-to-air and ethylene glycol-to-air types, as well as air-to-air applications such as between-refrigerants heat interchangers and straight air-to-air units. The fundamental principle of the inner-fin design is given special emphasis.
Dunham-Bush, Inc., West Hartford 10, Conn.

Reducing Valves. Of the pilot-operated type the pressure reducing valves of Bulletin J-170 function from line pressures to provide tight shut off, accurate control and minimal maintenance. Suitable for pressures to 125 psi and temperatures to 406 F. Information included offers installation, operation and specification aid.
OPW-Jordan, 6013 Wiehe Road, Cincinnati 13, Ohio.

Motor Application Guide. From 1/20 to 400 hp, polyphase, single phase and for direct current the motor offerings of this manufacturer are defined, illustrated, and reviewed thoroughly in all their design and operating aspects in Form 270 A, 16 pages.
Century Electric Co., St. Louis, Mo.

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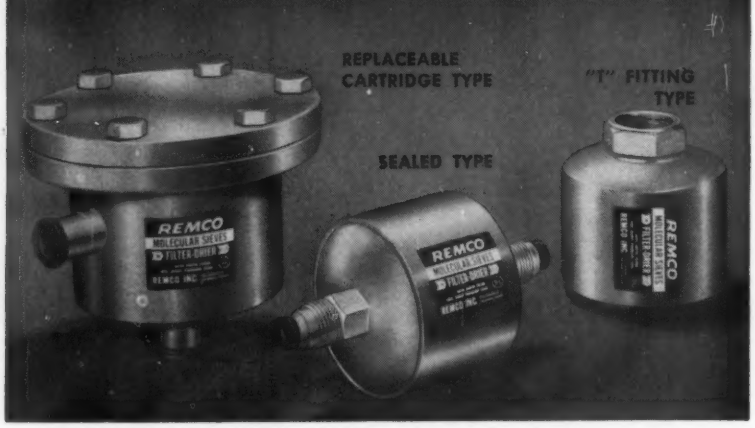

Utilizing advanced design Molecular Sieve cartridges, these new Remco Filter-Driers combine unequalled drying efficiency, effective acid removal, generous flow capacity and *depth* filtration.

The massive depth filter completely removes all scale, sludge, carbon and other particles as small as 100 microns. Molecular Sieves adsorb and retain large quantities of moisture even at refrigerant temperatures of 140F, and keep moisture concentrations below 10 ppm. Acids are reduced far below dangerous corrosion limits.

Compact in size, the filter-driers are U/L Approved and may be used for Refrigerants 12 or 22, Carrene or methyl chloride. Working pressure is 500 psi; minimum bursting pressure, 2500 psi.

REPLACEABLE CARTRIDGE TYPE units use an "O" ring for a positive, leakproof flange seal. From 3 to 40 tons, with ⅜" thru 1⅝" sweat connections.

SEALED TYPE filter-driers are available in 1 to 12 tons, with ¼" thru ⅝" flare and ⅜" thru ⅞" sweat connections.

"T" FITTING TYPE in 2 to 6 tons, are readily adaptable to systems using conventional "T" driers.

Remco Molecular Sieve Filter-Driers are available at leading wholesalers. Ask your wholesaler for more information, or write for Bulletin MS-1. Remco, Inc., Zelienople, Pa.

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Synchronous Motors. Designed for both commercial and industrial purposes, the constant speed motors reviewed in 4-page folder GEA-6814 are provided in open and dripproof, splashproof, weather protected, forced ventilated, totally enclosed (air-water cooled) and pressurized (inert gas) enclosures. Ratings are 50 to 1500 hp.
General Electric Company, Schenectady 5, N. Y.

Product Guide. Indexing the various offerings of this manufacturer, Bulletin G2-1a, within its 24 pages, covers a broad variety of other bulletins, catalogs and data sheets allied with specific instruments, controls, valves and auxiliary equipment.
Minneapolis-Honeywell Regulator Company, Industrial Div, Wayne & Windrim Avenues, Philadelphia 44, Pa.

Pneumatic Valves. Single seated and three way pneumatic coil valves as well as single and double seated diaphragm control valves are the subject of 4-page Folder 76-4591.
Minneapolis-Honeywell Regulator Company, 2747 Fourth Avenue, Minneapolis 8, Minn.

Spray Nozzles. Profusely illustrated to provide the specifier with specific details of spray and flow patterns obtainable with each of the nozzles defined, Catalog No. 59 is a 32-page introduction to a broad line of industrial spray facilities of this manufacturer.
Wm. Steinen Mfg. Co., Industrial Nozzle Div, 45 Bruen Street, Newark 5, N. J.

V-Belt Drives. Fully descriptive and definitive of a new line of V-Belt drives featuring compactness with cost savings, Bulletin A-695 covers a 1 to 1500 hp line with engineering thoroughness in its 44 pages.
Dodge Manufacturing Corporation, Mishawaka, Ind.

Circuit Breakers. Available in several series, each of which is delineated in an 8-page bulletin, these units can be used as combination switches and circuit breakers, are instantaneous tripping, are precision calibrated and are applicable to secondary voltages of dc and ac.
Wood Electric Company, 244 Broad Street, Lynn, Mass.

Cooling Towers. Bulletins 55-902 and 56-902 are summarizing guides, respectively, to hot-dipped galvanized redwood cooling towers of 5 to 100 ton capacity.
J. F. Pritchard & Co., 4625 Roanoke Parkway, Kansas City 12, Mo.

PARTS AND PRODUCTS

DUAL UNIT

Combining electric radiant heating with built-in lighting, this model unit is ceiling-recessed to save space. Rated at 1000 watt, the Pyronlite-RCL is mildly fan-forced to circulate convection heat normally stacked at the ceiling. Quick and even heating is achieved by tempered Pyrex glass panels having an electrically conductive coating. 200 watt of lighting are banked on either side of the heating panel.

Heating is controlled by a wall-mounted thermostat, lighting by a separate switch.

Berko Electric Manufacturing Corporation, 212-40 Jamaica Ave., Queens Village 28, N. Y.

RIGID FOAMS

Closed cell polyurethane foam, said to combine fire resistance with high strength and other properties necessary to foamed insulations, is being produced under the name Hetrofoam.

Hooker Chemical Corporation, Box 344, Niagara Falls, N. Y.

LINING-PADDING FELT

Non-woven felt, for padding of refrigeration equipment and other uses, is constructed when a multidirectional web of synthetic fibers is fed into needle looms of varying thicknesses, and the fibers interlocked. With certain fibers, or depending on the characteristics desired, the material is subjected to heat. This produces instantaneous shrinking, giving strength to the finished product.

Troy Blanket Mills, 200 Madison Ave., New York 16, N. Y.

PILOT CONTROL ADJUSTER

Positioning of the set point in this firm's indicating controllers by a 3 to 15 psi pneumatic signal is now possible with the addition of a special adjuster as an integral component. It is an optional feature.

U. S. Gauge Div, American Machine & Metals, Inc., Sellersville, Pa.

AMALGAM THERMOMETER

Temperature measurements of −57 C are considered possible with mercury-thallium filled thermometers developed specifically for low temperatures.

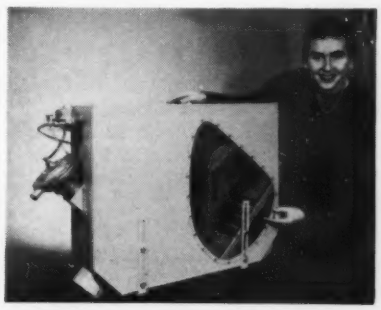
The mercury-thallium amalgam leaves the core clean, eliminating the formation of condensed droplets and chemical changes which may occur with organic liquid filled instruments. Reproducible readings at repeated intervals may be obtained under all climatic conditions.

In seven models (ranges of −57 to 100 C), the thermometers are furnished with yellow backs for clear readings.

H-B Instrument Company, American & Bristol Sts., Philadelphia 40, Pa.

ROOM TERMINALS

High velocity room terminals for all air systems include three models



which handle from 50 to 1700 cfm. The unit illustrated is for a double duct system. Other lines are single duct with reheat, and dual conduit.

These Weathermaster blending boxes offer versatility in arrangement of external controls, mounting rods and design. Features include the minimum distance (¾ in.) traveled by sliding damper blades, and the effective elimination of sounds from high pressure air.

Carrier Corporation, Syracuse 1, N.Y.

SPACE SAVING NOZZLE

For use in areas of limited space, this nozzle is designed with the whirl chamber at the back end and the orifice recessed into the nozzle body. This brings the apex of the spray cone flush with the mounting surface. Model CW7590 has a flow rate of ¾ gpm at 40 psi and a spray angle of 90 deg.

Bete Fog Nozzle, Inc., 309 Wells St., Greenfield, Mass.

WALK-IN TEST CHAMBER

Built to test missile components, this testing chamber has an altitude range from ambient to 100,000 ft, a temperature range from −100 to 300 F, and a relative humidity span from 20 to 95%. The 10 x 8 x 12 ft chamber requires a door weighing over three ton. This is suspended from a trolley and rail, to permit pushbutton operation.

Controls located inside the chamber, which govern the lowering of altitude to ambient, and which open the door, are an added safety feature.

American Research Corporation, Farmington, Conn.

DISPOSABLE FILTER

Parabolic in shape, this disposable bag filter is designed to have filtering action strongest where the dust particles are the finest.

The shape of the bag was determined on the principle that large particles of dirt in an air stream cannot change direction, due to inertia, and therefore will lodge at the back of the parabolic filter bag.

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Arco Manufacturing Corporation, 542 W. 55th St., New York 19, N. Y.

PACKAGED UNITS

Forty- and fifty-ton cooling capacity packaged air conditioning units have been added to this line for commercial and industrial applications. The complete range includes ten single compressor models rated from three to 50 ton.

The units use separate fan sections for vertical or overhung mounting with multiple fan discharges to fit any desired application. Units may be modified to make water connections on either end, and to locate the control panel on two alternate sides.

Units are with either three sizes of water-cooled condensers, or without a water-cooled condenser for air-cooled or evaporative condenser applications.

Westinghouse Electric Corporation, Air Conditioning Div, P. O. Box 510, Staunton, Va.

COMPRESSOR SURGE CONTROL

Fully electronic sensing, measurement and control instruments are features of this surge control system for large compressors, blowers and exhausters.

An electro-hydraulic or electro-pneumatic actuator and control valve is the final control element in the system, which is adaptable to every mode of compressor operation. Completely packaged, it may be used on units from 50 hp and up.

CDC Control Services, Inc., 419 S. Warminster Rd., Hatboro, Pa.

CONDENSERS

This line of low temperature condensing units now includes a 1½ hp size model, BRL15OTA, air-cooled, with an oil cooling coil. It uses Refrigerant-22.

Bendix-Westinghouse Automotive Air Brake Company, 950 East Virginia St., Evansville 11, Ind.

ASHRAE APPLICANTS

(*Continued from page 97*)

SIVERTS, HENRICK,* Asst. Appl. Engr., Minneapolis - Honeywell Regulator Co., Portland.

FOREIGN

Arabia

KILZI, GUY, Asst. Mech. Engr., Kuwait Oil Co., Kuwait, Arabia.

Australia

BECKMAN, LYALL, Refrig. Service Engr., Brisbane, Queensland, Australia.

Belgian Congo

NEEFS, JACQUES, Prop., ENELCA, Stanleyville, Belgian Congo.

Brazil

BERTOMEU, FRANCISCO, Tech. Dir., Industria de Refrigeracao Consul S. A., Joinville, Sta. Catarina.

England

FLIGHT, T. N., Tech. Dir., The British Anthracite Co., Ltd., St. James, London.

France

CARAYOL, A. P., Chief Engr., S. A. T. A. M., La Courneuve (Seine).

French Equatorial Africa

PICARD, ROGER, Tech. Mgr., A-C. Dept., Societe Comerciale de L'Ouest Africian, Abidjan, Ivory Coast.

Germany

VON CUBE, H. L., Gen. Mgr., Alfred Teves K. G., Frankfurt/Main.

Hong Kong

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India

THURKAL, C. B., Tech. Supvsr., 505 Army Base Workshops, Delhi.

Italy

BARBIERI, GAETANO, Joint-Prop., Costruzioni Meccaniche Barbieri, Bologna.

DE BERNARDI, CESARE, Exec. Vice-Pres., Aeromeccanica Star Corp., Turin.

DE DOMINICIS, GIOVANNI, Pres., Aeromeccanica Star Corp., Milan.

ROMEO, SCARIONI, Owner, Siltal, Abbiategrasso, Milan.

Okinawa

KWAN, SENG, Acting Mgr. & Proj. Engr., American Engineering Corp., Naha.

South Africa

OBORN, V. G., Sales Mgr., G. H. Langler & Co. Ltd., Durban.

RADFORD, R. P., Owner, R. P. Radford Refrig. Specialist, Sea View, Natal.

Sweden

PERSSON, P. O., Chief Engr., Helsingborgs Fryshus Aktiebolag, Stockholm.

Venezuela

ROMERA, C. R., Engr., EMCA Tecnia, C. A., Caracas.

SCHLAPFER, W. L.,* Chief Engr. A-C., Climasa S. A., Caracas.

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KILBURN, K. R., Student, University of Toronto, Toronto, Ont., Can.

KILBURN, M. C., Student, Purdue University, Lafayette, Ind.

KINGREY, R. R., Student, College of Texas A. & M., College Station, Texas.

KULESA, R. L., Student, Purdue University, Lafayette, Ind.

LANDON, G. E., Student, College of Texas A. & M., College Station, Texas.

LAWRENZ, O. A., Student, Agricultural & Technical Inst., Alfred, N. Y.

LEE, H. D., Student, University of Wisconsin, Madison, Wis.

LEIDOLF, E. J., JR., Student, College of Texas A. & M., College Station, Texas.

MAXWELL, A. H., Student, University of Toronto, Toronto, Ont., Can.

MCMILLAN, M. D., Student, Alabama Polytechnic Inst., Auburn, Ala.

MURRAY, H. S., JR., Student, Purdue University, Lafayette, Ind.

OGILVIE, A. F., Student, University of Toronto, Toronto, Ont., Can.

O'HIGGINS, P. J., Student, University of Toronto, Toronto, Ont., Can.

PARRETT, G. W., Student, University of Toronto, Toronto, Ont., Can.

PERKONS, EDUARDS, Student, Univer-

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sity of Toronto, Toronto, Ont., Can.
POLLEX, P. A., Student, University of Toronto, Toronto, Ont., Can.
PRITCHARD, M. E., Student, University of Toronto, Toronto, Ont., Can.
RICHTER, S. J., Student, University of Toronto, Toronto, Ont., Can.
SPURGEON, S. J., JR., Student, Alabama Polytechnic Inst., Auburn, Ala.
TARASUK, J. D., Student, University of Toronto, Toronto, Ont., Can.
THOMPSON, B. H., Student, College of Texas A. & M., College Station, Texas.
THORNTON, A. B., Student, University of Toronto, Toronto, Ont., Can.
TUCKER, H. W., Student, Alabama Polytechnic Inst., Auburn, Ala.
THACKSTON, P. L., Student, North Carolina State College, Raleigh, N. C.
WATSON, H. L., III, Student, North Carolina State College, Raleigh, N. C.
WILSON, C. C., Student, University of Toronto, Toronto, Ont., Can.
WORSLEY, C. E., Student, University of Michigan, Ann Arbor, Mich.

PARTS AND PRODUCTS

DIAL THERMOMETER

Designed to meet the requirements of all types of frozen food cabinets and cases, this instrument, as illustrated, has a 2½-in. dial with an increasing scale which provides wide spacings in the reading sector.

Accurate to plus or minus one division over its entire range of −20 to 60 F, the thermometer is of the Bourdon tube vapor tension type. It is standard with 5½ ft of nickel plated capillary tubing and a 3½-in. stem.
James P. Marsh Corp., Skokie, Ill.

HEAT TRANSFER METER

Designed for use in measuring heat transfer in hot water generators, cooling towers, refrigerating systems and other such applications, this Btu meter is a single recorder which continuously records, indicates and integrates Btu loss or gain in a flowing liquid.

Temperature differential and flow rate of the liquid are also recorded.

Two platinum resistance temperature elements, differential connected and measuring inlet and outlet temperature of the liquid, form two legs of a resistance bridge in which the differential temperature receiver in the recorder acts as a rebalancing unit.

Liquid flow through the process, measured by a flow transmitter, is also recorded. Differential temperature and flow retransmitting slidewires on their respective receiver units are connected so as to multiply their respective signals. This product represents number of Btu's gained or lost per unit of time by the liquid as it passes through the process. The Btu signal is directed to the Btu receiver unit, where it is rebalanced and recorded.
Hays Corporation, Michigan City, Ind.

VINYL-COATED STEEL

Plastics and steel sheet have been combined to create a product with

applications ranging from portable electric heaters and room coolers (as in the illustration) to picnic coolers.

The production of this vinyl coated steel sheet involves the curing and bonding of liquid vinyl plastisols to sheet steel in a continuous mill coating process. In many textures, it may be embossed (prior to cooling) with any texture that can be engraved on a printing roll, and produced in any specified color with uniformity.

Coatings of the liquid vinyl range from 0.008 to 0.020 in. thick. The steel sheets are in gages from 18 to 28. Widths range from 24 to 52 in., lengths from 30 to 144 in. It may also be supplied in coils.

The new material is considered heat resistant up to 160 F on a continuous basis (under 212 F for two days). It does not support combustion.
United States Steel Corporation, 525 William Penn Place, Pittsburgh 30, Pa.

THERMISTOR SENSING DEVICES

Highly flexible, compact and simple in operation, this temperature monitoring system is designed to detect and furnish warning when the temperature at any protected point reaches a preselected high or low limit. The system, a continuous one, may be used for monitoring temperatures between −25 and 600 F.

Scanning techniques are not needed; the sensing points are terminated in a multiple-channel bridge network which receives and monitors all inputs simultaneously and continuously.

It is designed around two types of modular units: a monitor module, which handles the inputs from a

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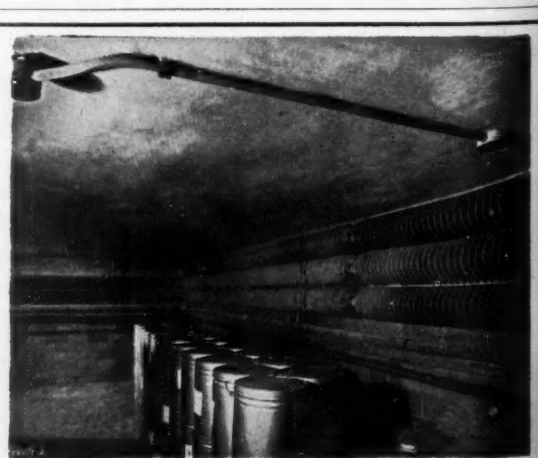
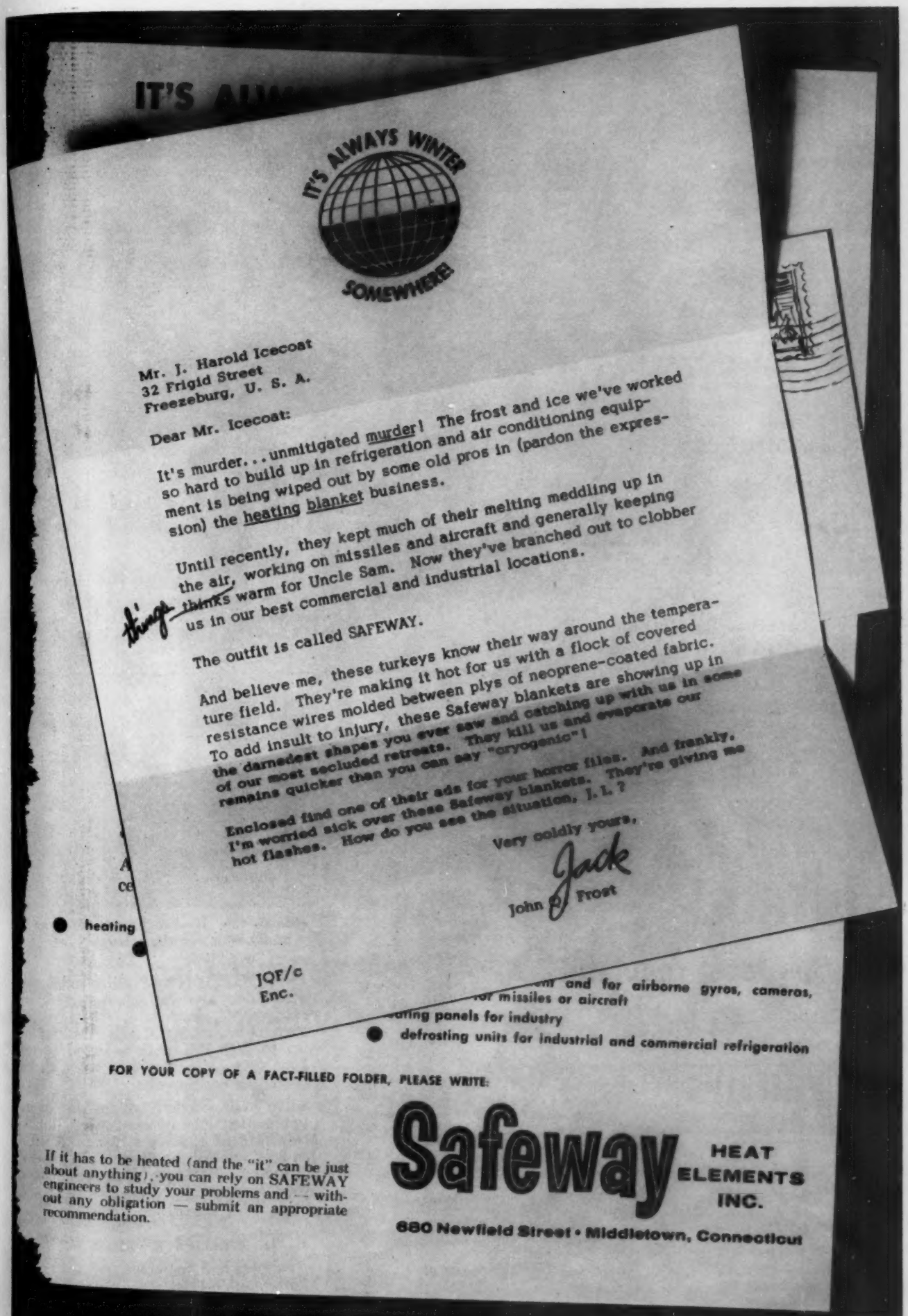


DIMCO Fin Coils Solve Low Headroom Problem In a Mezzanine Lard Storage Room


IT'S ALWAYS WINTER SOMEWHERE!
Mr. J. Harold Icecoat
32 Frigid Street
Freezeburg, U. S. A.
Dear Mr. Icecoat:
It's murder...unmitigated murder! The frost and ice we've worked so hard to build up in refrigeration and air conditioning equipment is being wiped out by some old pros in (pardon the expression) the heating blanket business.
Until recently, they kept much of their melting meddling up in the air, working on missiles and aircraft and generally keeping things warm for Uncle Sam. Now they've branched out to clobber us in our best commercial and industrial locations.
The outfit is called SAFEWAY.
And believe me, these turkeys know their way around the temperature field. They're making it hot for us with a flock of covered resistance wires molded between plys of neoprene-coated fabric. To add insult to injury, these Safeway blankets are showing up in the darnedest shapes you ever saw and catching up with us in some of our most secluded retreats. They kill us and evaporate our remains quicker than you can say "cryogenic"!
Enclosed find one of their ads for your horror files. And frankly, I'm worried sick over these Safeway blankets. They're giving me hot flashes. How do you see the situation, J. L.?
Very coldly yours,
John Frost
JQF/c
Enc.
heating
and for airborne gyros, cameras,
for missiles or aircraft
defrosting panels for industry
defrosting units for industrial and commercial refrigeration
FOR YOUR COPY OF A FACT-FILLED FOLDER, PLEASE WRITE:
If it has to be heated (and the "it" can be just about anything), you can rely on SAFEWAY engineers to study your problems and — without any obligation — submit an appropriate recommendation.
Safeway HEAT ELEMENTS INC.
880 Newfield Street • Middletown, Connecticut

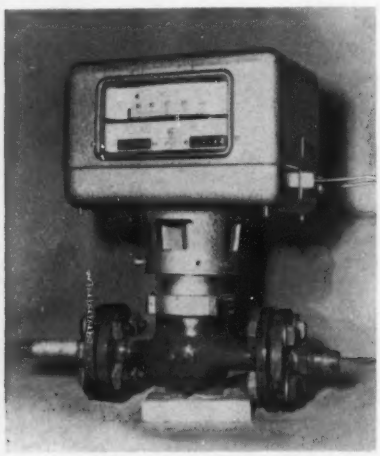
group of 10 sensing points; and an alarm-indicator module, which contains an amplifier, temperature indicator, alarm circuit and connections for any secondary switching required.

Various operating ranges of 100 F are available within the limits cited above.

Fenwal Inc., Ashland, Mass.

HEAT PUMP METER

For use with the heat pump, this Btu meter measures both heating and



cooling transfers in commercial installations.

This makes it possible to charge tenants for both heating and cooling from a central heat pump installation through a single meter, in exact proportion to what each tenant uses.

The device consists of a liquid meter, an integrator and two temperature sensing bulbs. One bulb is placed in the supply line and the other in the return line. The difference in water temperature between supply and return line is multiplied mechanically by the integrator with the flow of water which has been converted to net weight. The result in Btu is shown cumulatively on a digetal counter on the face of the meter.

One dial shows heating, a second cooling, depending on what cycle is in use. Instantaneous temperature is also shown.

Air Conditioning Equipment Corporation, New York, N. Y.

IMPROVED CONTAINERS

Ring-top refrigerant cylinders made for packaging this firm's brand of synthetic refrigerants feature a steel shield which will also act as an auxiliary stand.

The cylindrical ring-top steel shield on hand-carried 10 and 25 lb cylinders is used for valve protection rather than as a cap. By turning the unit upside down, it becomes a base for support. Cut-away sections in the shield form two handles for lifting and carrying the cylinder.

Union Carbide Chemicals Company, 30 E. 42nd St., New York 17, N. Y.

MOTORIZED PUMP

Small cooling tower and chilled water applications are the specific function of this direct-connected centrifugal pump of all bronze construction.

With either ¼ hp or 1/3 hp, 3450 rpm 110 volt or 220 volt, 60 cycle, 1 phase motor, the pump has a 1¼-in. suction connection and a 1-in. discharge connection.

The 1/3 hp pump capacity varies from 40 gpm of water at 22 ft head to 2½ gpm at a 34 ft head. The direct connected impeller eliminates bearings in the pump so that the entire assembly is only slightly larger than the motor.

A ⅛-in. pipe plug in the cover plate permits draining and air purging. The pump, under the trade name Cool-Flo, may be rotated to any discharge position by loosening latch-bolts on the motor.

Thermalair Engineering Company, 7535 W. Eight Mile Rd., Detroit 21, Mich.

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IN ADDITION — Phillips pilot operated valves are available for all common refrigerants, down to —50° F. Operates with as low as 2 PSI pressure drop and up to 250 PSI with selected springs. Line sizes ½ inch to 4 inches with steel or copper connections.

Solve your flooded system design and application jobs by consulting Phillips. Our firm of engineers have specialized in level control, liquid-vapor separation, liquid circulation and return systems for over 28 years.

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Designers and Engineers
Refrigeration Control Systems
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Chicago 24, Illinois

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In addition, your customers will benefit from greatly simplified motor maintenance. Pre-lubrication coupled with fortified insulation and close-tolerance machining makes the Life-Line "A" the nearest thing to a maintenance-free motor available.

For more information on the Life-Line "A" for air conditioning drives, contact your Westinghouse representative. Or, write Westinghouse Electric Corporation, P.O. Box 868, 3 Gateway Center, Pittsburgh 30, Pa. J-22127

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ASHRAE Activities Defined

Here, in summary, are some of those specific things which define the American Society of Heating, Refrigerating and Air-Conditioning Engineers—

Fields. Engineering aspects of heating, refrigerating, air conditioning and ventilation as allied with research, development, installation, equipment, standards and industry cooperation.

Membership. Approximately 17,800 members. Initially 84 local chapters organized within 10 geographical regions.

General and Special Committees. Covering the full range of society, technical, standards and inter-society activities. See page 90 of this issue for an early listing of General and Special Committees.

Research. Continuation and augmentation of the research program conducted at the Society's Cleveland Laboratory with supplementary grants to independent educational and investigatory institutions.

Standards. Continuation and amplification of the Society's work on engineering standards with projection into various phases of heating, refrigerating, air conditioning and ventilation.

Publications. The several publishing activities of the merged societies will be as follows:

ASHRAE Journal. The monthly, engineering, official publication of the Society reporting upon national meetings papers, outstanding developments reported independently, news of Chapters, engineering news of the industry and happenings related to members as well as significant developments in the component products, parts and materials useful to them in their work.

Transactions. Complete annual reporting upon the activities of the Society. Technical program, social and official business reports.

Guide and Data Books. Encyclopedic handbook presentations of authoritative information covering fundamentals, applications and equipment in the fields of heating, refrigerating, air conditioning and ventilation and issued in annual volumes upon a rotating program.

Standards. Published, as developed and approved, for distribution as requested.

Research Reports. Recording the outcome of Laboratory and sponsored projects.

Meeting Preprints. Distributed at national meetings to facilitate study and discussion of papers presented at technical sessions.

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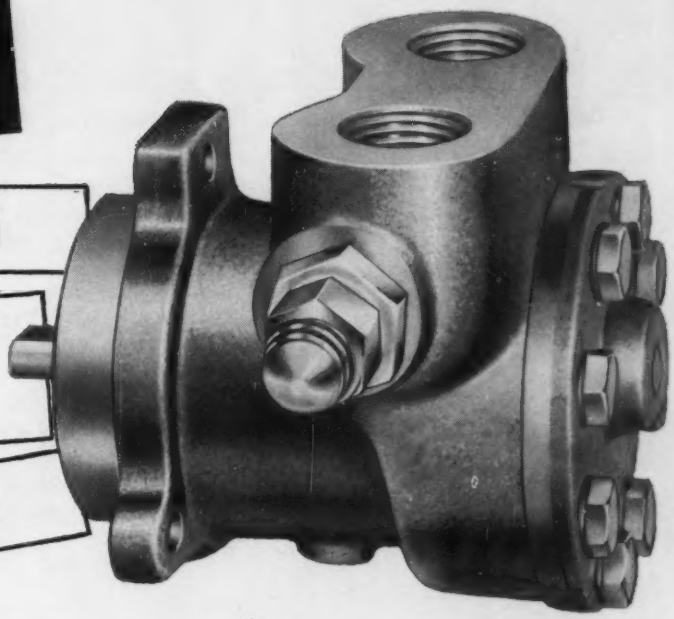
Complete mounting interchangeability

Built-in relief valves



on TUTHILL small industrial pumps

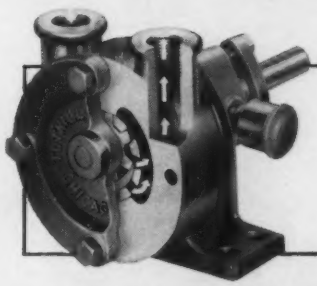
Now a new design—Series LPFV—gives you even greater flexibility with Tuthill small industrial pumps . . . the industry standard for dependable operation for many years.

Complete interchangeability allows the use of any one of five different sizes in the series on the same mounting arrangement. Capacities range from 20 to 360 gallons per hour, at pressures up to 500 psi. Normally provided for 1800 RPM, 3600 RPM speeds are permissible in most cases. And a built-in relief valve is now offered as optional equipment on all LPF models.

Compactness, an outstanding characteristic of all Tuthill pumps, is particularly exemplified in the new model LPFV with its shorter mounting hub. It is also available in close coupled motor-pump combinations for applications where space and weight are at a premium.

The newly designed model LPFV incorporates the performance-tested operating characteristics of Tuthill's internal gear construction described at right. In thousands of applications . . . in hydraulics, lubricating, transfer, circulating and other services . . . Tuthill internal gear pumps have established enviable records for reliability and quiet operation.

Over 700 models of Tuthill internal gear pumps are available to provide one especially suited to your application. These include stripped models for built-in applications, cartridge pumps, reversible models . . . a host of specialized pumps for each individual service. Mail the coupon today.



TUTHILL internal gear pumping principle

In Tuthill internal gear pumps, proven in 30 years of operation, there are only two moving parts. The principle is based upon the use of a rotor, idler gear and a crescent shape partition cast integral with the cover. Power applied to the rotor is transmitted to the idler gear with which it meshes. The space between the outside diameter of the idler and the outside diameter of the rotor is sealed by the crescent. As the pump starts the teeth come out of mesh increasing the volume. This creates a partial vacuum drawing the liquid into the pump through the suction port. The liquid fills the spaces between the teeth of the idler and the rotor and is carried past the crescent partition to the pressure side of the pump. When the teeth mesh on the pressure side the liquid is forced from the spaces and out through the discharge port.

Tuthill Pump Company
969 East 95th Street
Chicago 19, Illinois

☐ Please forward information on LPF series
☐ Please have your representative call

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Write R. C. Sherer, Vice President . . . an area engineer is at your service.

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precision temperature controls for what's doing today . . . and coming tomorrow.

With ASHRAE Chapters

(*Continued from page* 95)

sign Features in Modern Compressors was presented by Joseph Raufeisen; and Norman Sharp concluded with his address on the New Answers to Lubrication Problems on Refrigerant 12 and 22 Systems.

A talk on the application of ammonia entitled Do We Always Do the Best Way, was presented at the February meeting. Charles R. Kohlenberger, vice president of Kohlenberger Engineering Corporation commented on the use of high side equipment, and Robert S. Ohling, project engineer with the same firm, spoke of the low side equipment and applications.

Low first cost and efficient plant operation dictate the use of ammonia as the refrigerant in most storage and freezer plant systems, it was noted. Both speakers cautioned against undersizing condensers and evaporators to save on first cost, as it will not be economical in the long run. Use of evaporator surfaces, lower temperature differentials and face velocities of 800 to 1200 fmp help, they maintained, to reduce the defrost problem.

EVANSVILLE . . . Planned for the February meeting of this combined chapter was a lecture on Solid Nuclear Fuels for Military and Civilian Use. James J. Dutton, Assistant Manager of Marketing and Contract Administration, M & C Nuclear, was the speaker.

ONTARIO (H) . . . Combustion of Natural Gas in Boilers was up for discussion in March when L. S. Reagen, President of Webster Engineering Company, addressed the members.

ARIZONA (H) . . . Clay Snider, original regional sales manager for Minneapolis-Honeywell Regulator Company, presented a talk on the Economic Advantages of Clean Air, at a recent meeting.

NATIONAL CAPITAL (R) . . . Paul Achenbach of the National Bureau of Standards planned to meet with chapter members in March to cover air, moisture and heat transfer in refrigeration structures.

SAN FRANCISCO (R) . . . On the agenda for the February meeting of this chapter was a talk on the Application of Suction Pressure Regulators for Refrigeration and Air Conditioning. Bill Baker, Baker Company, was guest speaker.

JACKSONVILLE (H) . . . Spurring an informative question and answer period, Dave Lawson, Sporlan Valve Company, addressed the group in February on Refrigeration Control and its Problems.

BLUEGRASS (H) . . . Planning to cover the basic concept, function and development of radiant conditioning, Pat Malone, Burgess-Manning Company, appeared before the March meeting of this chapter.

MEMPHIS (H) . . . Expansion valve selection, application and function were topics at the Feb-

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ruary meeting when W. J. Franklin, Alco Valve Company, presented a program on Refrigerant Controls.

MILWAUKEE (R) . . . In March, chapter members were to hear Professor W. F. Stoecker, University of Illinois, speak on Ammonia Recirculation and Return Systems. He is Assistant Professor of Mechanical Engineering.

PUGET SOUND (H) . . . Panel members R. H. Schairer and Gilbert Mandeville presented their views on the engineering of the air conditioning systems in Seattle's new office buildings. Donald Owens and Warren Ludwig gave theirs on the equipment used, when members gathered in February.

ROCKY MOUNTAIN-COLORADO . . . Members of both local chapters, meeting jointly in February, heard F. A. Grossman, Martin Company, on Rockets, specifically, the work being done by his firm on the Titan ICBM project.

WESTERN NEW YORK (H) . . . School heating with electricity and gas was discussed in its various aspects in February by William B. Campbell, Power Sales Supervisor, Niagara Mohawk Corporation, and Richard Duttweiler, Sales Manager, Iroquois Gas Company.

SOUTH CAROLINA (H) . . . February guest speaker Col. Edgar L. Morris, attorney and registered engineer, chose as his subject: the legal position of the engineer and his relationship with the architect, contractor and owner.

ST. LOUIS (R) . . . Centrifugal Compressors were on the agenda in February when these members invited H. C. Grubb, American Blower Div, American Standard Sanitary Corporation, to speak.

MOBILE (R) . . . Touring the Eglin AF Base in February, these members inspected the climatic hanger, test laboratory, the Bomarc Missile launching base and the power station which utilizes a high temperature hot water system.

WESTERN MICHIGAN (R) . . . Present Methods and Trends in Insulation in the Appliance Industry were covered when E. D. Cox, Insulation Department, Johns-Manville Company, presented an illustrated talk.

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VALVES AND ACCESSORIES FOR REFRIGERATION AND INDUSTRIAL APPLICATIONS

Applications

MOBILE TRACKING STATION SIGHTS MISSILES AT SEA

Eleven air conditioning systems remove the quantities of heat generated by electronic equipment on the **S. S. American Mariner,** a floating missile research station.

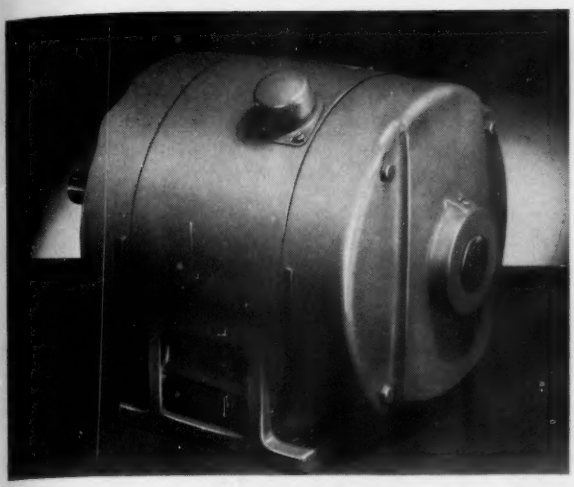
Patroling the South Atlantic, the refitted cargo vessel carries radar and optical apparatus for providing the most precise data on missiles fired by the United States military services.

Of the total of 285 ton provided by four reciprocating refrigeration machines, two of the systems use 55 ton to cool the electronic equipment directly. Carrier Corporation supplied the air conditioning plants.

INSURANCE DATA PROCESSING COOLING-LIGHTING COMBINED

To offset the heat from electronic data processing equipment installed in the Travelers Insurance Company offices in Hartford, Conn., the ceiling light fixture is combined with an air diffuser.

The draft-free air cooling method, the Multi-Vent Troffer, as designed by the Pyle-National Company



General Electric announces . . .

inherent protection for integral-hp three-phase motors for fan, blower, compressor applications

U.L. listed, G.E.'s inherent protection for integral-hp three-phase motors is long-awaited breakthrough in guarding against all abnormal operating conditions

You can now get a 1, 1½, 2 or 3-hp, 1800-rpm, three-phase motor with U.L. listed inherent protection that will *stay on the job*, delivering its maximum rated output, without danger of burnout.

General Electric's inherently-protected three-phase motor does a *complete protection job* by providing "built-in" protection against motor overheating under *all* abnormal operating conditions.

Engineers at General Electric's Small Integral Motor Department expect that "built-in" protection will virtually eliminate three-phase motor burnout caused by abnormal operating conditions, such as unbalanced voltage, single-phasing, prolonged overloads, stalling, failure to start, lack of ventilation, and ambient temperature increases.

SENSES ALL FACTORS

General Electric has achieved inherent protection for three-phase motors by mounting a protector near the motor windings, where it is sensitive to all factors that cause overheating. The protector is responsive to both temperature and current. This means that inherent motor protection is superior to external-control protection, since controls only protect against excessive motor current.

Nuisance trips or burnouts are eliminated by the protector, because it is calibrated to trip and open the motor circuit at the maximum safe winding temperature level. Here's how General Electric's inherent protection works:

TAMPER-PROOF ASSEMBLY

The three-phase motor protector has a snap-acting thermal disc, three contacts and three heating elements in a molded phenolic base. The unit is assembled on the motor frame, and protected by a pressed-steel cap. Internal connections are factory made at the neutral point of the three-phase wye-connected motor. (This assembly makes it tamper-proof, a further assurance of permanent motor protection.) An etched, stainless-steel plate, located on the motor frame near the conduit box, provides complete external connection information.

Each of the three protector contacts, and its associated heating element, is electrically in series with the neutral point of the wye-connection and a phase of the motor winding. When the thermal disc snaps open—either from high ambient temperature or from excessive motor current—connection is broken at the neutral point, interrupting the current in all three phases of the motor. The protector resets *automatically* after the motor cools, and motor operation is resumed with "built-in" protection.

EASILY MAINTAINED

In addition to the significant advance in three-phase motor protection offered by General Electric, the G-E inherently protected three-phase motor has been ideally designed for easy inspection and maintenance. The motor can be disassembled—end shields, baffles, rotor assembly, and bearings can be removed —without disturbing the protector or making time-consuming disconnections.

Manufacturers and users of fans, blowers and compressors can get more information on G.E.'s three-phase motors with "built-in" protection by contacting any General Electric Apparatus Sales Office. Or, information can be obtained from Section 840-27, General Electric Company, Schenectady 5, N. Y. (Ask for A-6932.)

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incorporates a flush-mounted fluorescent lighting fixture that hides a low velocity air diffuser attached immediately above. Eliminating the cluttered ceiling appearance, it controls the heat from the more than 20,000 electronic tubes in the processing units of the RCA Bizmac system.

INDOOR ICE RINK FEATURES ACCESSORY HUMP CONTROL

Addition of a Trane climate changer at each end of the indoor ice rink at the Golden Valley Ice Center in Minneapolis, Minn., resulted in controlling the relative high humidity and ice fog to eliminate condensation from roof trusses and the formation of those humps which characteristically occur where such control is not assured.

The rink is 185 x 85 ft. The structure seats 700 spectators. There are 9½ miles of 1¼-in pipe laid on 4-in centers throughout the rink area to carry the brine which freezes the ice. Different ice hardnesses are provided: a surface temperature of 25 F for hockey games, 30 F for recreational skating sessions. Two Trane cold generators provide the cooling required. Each is driven by a 100-hp motor and is capable of cooling 478 gpm of 25 per cent brine solution from 17 to 15 F when operating at 105 F condensing temperatures. Brine pumps circulate 1000 gpm against a 60-ft head.

POTATO CHIP MANUFACTURER HAS AIR CONDITIONED PLANT

With the exception of warehouse areas and cooking rooms, the entire plant of Gordon Foods Div of Sunshine Biscuits, Inc., in Atlanta, Ga., is air conditioned. This is accomplished through air units and distributing duct work with chilled water coils for cooling and steam coils for heating. The office areas use air units with combination hot and chilled water coils for the general areas and remote room type units for individual offices.

All of the plant area duct work has been kept as straight forward as possible, using side outlet grilles with an average throw of 30 ft. Since much of the cooling in the plant areas is designed primarily to reduce the heat in the product before packaging, perfect air distribution is not a problem. The number of people in the area is relatively small and they are normally doing light work. The duct work system in the office areas is the conventional diffuser type designed for comfort conditioning.

The chilled water system consists of two closed type centrifugal compressors, each with a capacity of 540 gpm when cooling water from 55 to 45 F. This water is circulated by two double suction pumps with a capacity of 540 gpm each. Condenser water is cooled by a double flow induced draft water tower.

Operation of the centrifugal compressors is automatic, with all units interlocked for safety. The units are operated in parallel with a pneumatic control system regulating chilled water temperature.

Steam for the air units is provided by two 300 hp package boilers, firing gas or light oil and operating at 100 psi. Approximately half of the steam load is

TRADE FRICK MARK®

Refrigeration

Air View Showing Five Buildings of the Chattanooga Food Terminal. Cold Storage has Cooling Tower on the Roof.

Serves Chattanooga Food Terminal

The new cold storage building at this terminal operated by Chattanooga Warehouse & Cold Storage Co. measures 660 ft. long by 100 ft. wide by 23½ ft. high, and is divided into eleven rooms. Eight rooms are convertible as freezer storages. The three freezer storage rooms include two air-blast quick-freezing tunnels. Complete refrigeration is supplied by a 3-temperature Frick System.

For that all-important cooling job—whether air conditioning, ice making, cold storage, food freezing or process work—look to Frick engineers for the last word in reliability.

● ***Write now for bulletins and estimates on your particular cooling requirements . . .***

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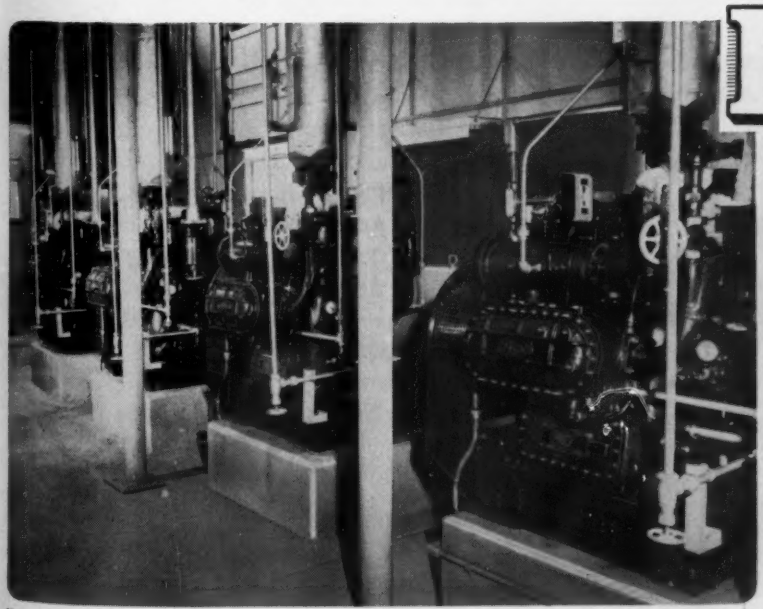
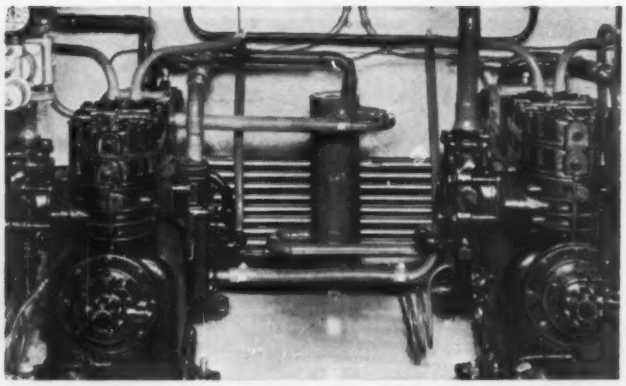
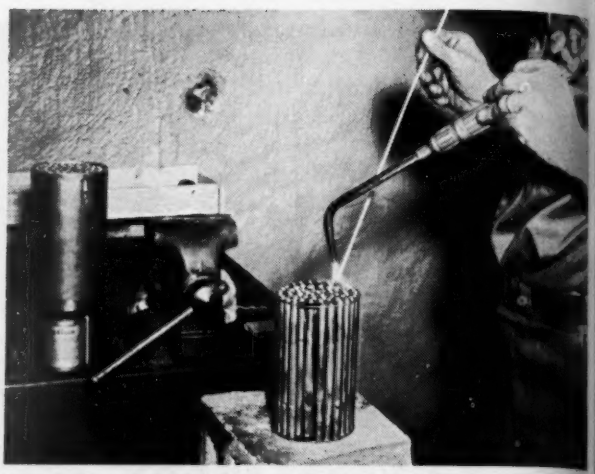


Four "ECLIPSE" Compressors Handling High-pressure Ammonia at Chattanooga Food Terminal.

Typical View Inside Cold Storage, with Air Cooling Unit above Doorway, at the Terminal.

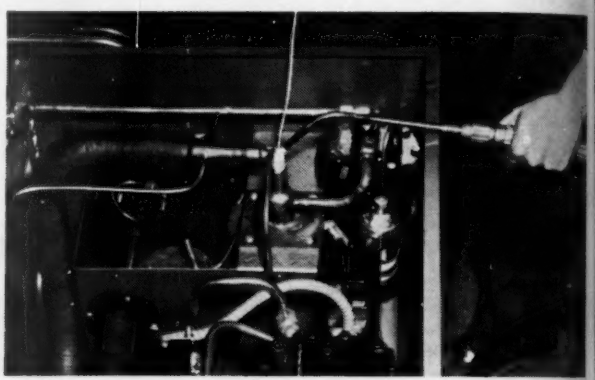


Here is a cascade refrigeration system using Freon 22 and Ethane refrigerant. This system cools five gallons of oil per minute to minus 70° F. All of the refrigerator tubing on both high- and low-pressure sides is brazed with SIL-FOS (approximately 185 joints). Some joints fluctuate in temperature from minus 200° F to plus 70 or 80° F.



Here, operator brazes the tube bundle before assembling it in condenser shell. Hand gas-air torch keeps the SIL-FOS flowing freely.

Now the tube bundle is being brazed to the sheet by the same simple heating method.



Operator puts finishing touches on last joints of small two-stage unit.

20 Years Without A Joint Failure... Harris Refrigeration Sets This Record With SIL-FOS

This 20 year failure-free record is even more remarkable since the joints of Harris equipment are subjected to extreme temperature, pressure and thermal shock far in excess of an ordinary refrigerator.

A pioneer in the design and manufacture of low-temperature refrigeration equipment for science and industry, Harris Refrigeration Company, Cambridge, Massachusetts, finds Handy & Harman SIL-FOS the brazing alloy that meets all its stringent requirements for high strength, leak-tightness and durability. Specified for 20 years in the production of Harris units ranging from small refrigerators for cold-treating ball bearings to a giant big enough to chill a 20-foot, 10-ton mill roll, SIL-FOS has always assured highest quality joints while providing the production flexibility and economy essential to efficient manufacturing.

Time, strength and leak-tightness are, by themselves, excellent reasons for specifying SIL-FOS. There are many more. Like brazing's inherent economy. Like the ease and speed with which joints are made. And like the quality SIL-FOS joints assure your product. We would like to give you all the necessary brazing background you may require ... call us.

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used for process and domestic water heating. Usually, the steam is put to 50 psi for use in the air units.

A system of pneumatic controls is used throughout the building, with air being provided from the plant air system. Seven air reducing stations are used, being centrally located to minimize the piping. A typical control system operates as follows:

The face damper over the cooling coil is normally open, bypass damper closed and steam valve closed. On falling temperature the control air pressure starts to increase, which in turn starts to close the face damper. At 8 psi the face damper is 100% closed and the bypass 100% open. At 7 psi the steam valve starts to open and at 11 psi it is completely open. On rising temperature the operation is reversed. This makes for a simple, accurate system and eliminates the problem of switching from summer to winter operation and vice versa.

In the air conditioning and ventilation of the potato storage rooms, where 2½ million lb of potatoes can be stored, the primary problem was one of ventilation. Air was distributed at ceiling level through diffusers directing over the wide aisle at the center of the room. The return air was picked up 18 in. above the floor and along the entire outside wall area. The system is balanced by means of an adjustable slot running the length of the return air duct.

Although it is impractical from a cost standpoint to air condition the cooking rooms, a certain amount of comfort is provided for the personnel there by taking the amount of outside air that is introduced for ventilation in the air conditioned packaging rooms and blowing it into the cooking rooms.

RIGID URETHANE INSULATES 38-FT TRUCK TRAILER

Fully insulated with plywood panels cored with four inches of rigid urethane foam, one of a group of five 38-ft refrigerated transport trailers was so equipped in about one quarter the time required for the material and method used for the other four.

Tests on heat loss conducted after a month of road service showed 2.6 Btu/hr/F/ft length for the trailer that was foam-insulated as compared with 3.3 Btu/hr/F/ft length for the other four. As supplied by the National Aniline Div of Allied Chemical, these urethane-filled panels have low moisture pickup and serve to temperatures of 250 F without deterioration.

STEAM-POWERED TURBINE FOR NEWS PRINTING

Air conditioning equipment totaling 1600 ton will be the nucleus of the system for the new New York Times subsidiary printing and distribution building. Two Trane Company Turbine CenTraVac machines, rated at 800 ton each, are driven by a steam powered turbine instead of the conventional electrical motor.

In this design, the impeller is mounted directly on the turbine shaft and is supported by the turbine bearings. An integral assembly is formed by the compressor and turbine. Just the ring collar of the modified shaft seal rotates.

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